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TWIN DISC GEAR TOOTH SIMULATOR

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This report describes the results of an effort to develop a disc on disc test rig for evaluating lubricant load capacity. The goal of the program has been to develop a reliable disc on disc (or Twin Disc) test rig capable of providing more reliable and lower cost evaluation of lubricant load capacity than the Ryder gear test which is currently used.

Disc rigs have been evaluated for this application in the past, but have failed to provide scuffing results which are comparable to those found in gear tests. The unique feature of the rig designed and evaluated under this program is a drive system design which varies the disc to disc sliding and rolling speeds in a fixed manner to simulate the combined rolling and sliding motion found in a gear tooth contact. Other features include a drive system design which insures that the same points on each disc always contact each other as they rotate, and material selection and heating system design for operation with experimental lubricants at temperatures up to 700°F.

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The drive system and overall rig design were developed using sinusoidal gears to simulate as closely as possible the combined rolling and sliding motion in a gear contact. The rig conditions are set to meet or exceed the maximum operating conditions found in the Ryder gear rig including such factors as average rolling speed, maximum sliding speed, maximum Hertzian stress, contact Blok flash temperature, and lubricant inlet temperature.

Test results have shown that scuffing does occur on the disc samples which is similar to that found in gears, good reproducibility is found for repeated tests on the same lubricant, disc samples are less expensive and easier to manufacture than test gears, and test discs can be reground and reused for successive tests reducing further the cost of the test samples. Differences have been found in the relative ranking of test lubricants which can be explained by the detailed differences in the bulk disc temperature versus the contact flash temperature compared to the bulk Ryder gear temperature versus its flash contact temperature when testing the same lubricant. Differences found are particularly noticeable with unformulated lubricants. Development of a detailed understanding of these phenomena should permit the rig to be used to evaluate either lubricants or gear materials for their potential to perform adequately in a gear application.

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ABBREVIATIONS, ACRONYMS, AND SYMBOLS

BE	Tooth curvature radius, in.
C	Gear material specific heat, Btu/lb/°F
C_1	Ratio at start of transition
C_2	1/2 (ratio at end of transition - ratio at start of transition)
C_3	Scale factor such that $C3 \times \theta = 180°$
E	Gear material bulk modulus, lb/in.
E_1	Gear material bulk modulus, lb/in., for disc 1
E_2	Gear material bulk modulus, lb/in., for disc 2
E_r	Elasticity factor
EHD	Elastohydrodynamic
M_p	Contact ratio
N	Number of teeth
N_D	Number of driver gear teeth
N_S	Number of slave gear teeth
P	Diametral pitch
PV	Pressure-velocity
R	Radius, in.
R_1, R_2	Instantaneous gear tooth radii at mesh, in. (Section 2.2) Radius at contact, in. (Section 2.5)
R_b	Base circle radius, in.
R_g	Outer gear radius, in.
R_p	Pitch circle radius, in.
RC	Ratio charge
T_f	Critical flash temperature, °F
V_1	Driver gear tooth tangential velocity, in./sec
V_2	Slave gear tooth tangential velocity, in./sec
V_r	Rolling velocity, in./sec
V_s	Sliding velocity, in/sec
W	Unit load, lb/in.
Wt	Unit load at maximum test load, ln/in.
f	Friction coefficient (Section 2.4) Shape factor (Section 2.5)
h	Film thickness, in.
k	Thermal conductivity, lb/°F sec
α	Angular acceleration, sec^{-2}
β	Driven gear starting angle, degrees (Section 2.2) Thermal contact coefficient (Section 2.4)

θ	Input angle, degrees
ν	Poisson ratio of gear material
ν_1	Poisson ratio of disc 1 (Section 2.5)
ν_2	Poisson ratio of disc 2 (Section 2.5)
ξ	Driver gear starting angle, degrees
ρ	Density, lb/in.^{3}
σ_{Hz}	Hertzian contact stress, lb/in.^{2}
ϕ	Pressure (contact) angle, degrees
ω	Angular velocity, radians/sec

1.0 INTRODUCTION AND BACKGROUND

Increased requirements imposed on modern gas turbine systems result in higher turbine speeds with increased loading of bearings and gears. Associated with this demand for increased performance is the heightened demand on the turbine lubricant to adequately perform its various designed functions. Included in these functions is the ability to effectively carry imposed operating loads, thus minimizing metal-to-metal contact.

Lubricant load capacity has, therefore, become a very important consideration in the design of gas turbine systems. For this reason, testers that can be used to evaluate the performance of lubricants and gear materials under conditions similar to those between meshed gear teeth are important in guiding the development of improved compositions of each. They are also an essential tool in controlling lubricant and gear material quality through material specifications.

1.1 Statement of Problem

Currently, the Air Force follows ASTM-D-1947, "Standard Test Method for Load Carrying Capacity of Petroleum Oil and Synthetic Fluid Gear Lubricants," to rate the gear load-carrying capacity of candidate turbine lubricants. This test method specifies the use of the Ryder Gear Tester, which uses standardized gears that are loaded together in a four-square pattern by torquing the rotating assembly. Both the gear load and the gear speed are test variables. The Ryder tester has had wide usage by both lubricant suppliers and users. The accumulated volume of test data gathered from these many sources provides a base for judging lubricant load capacity.

Over the years, however, problems related to the use of the Ryder tester have been reported with increasing frequency. One problem has been the increased cost and reduced vendor availability for the test gears themselves; the second has been the variability in the test data. This variability may be the result of operator interpretation and/or may be due to the variance of both test gears and lubricants [1, 2]. Because of the data variability, numerous tests must be run to establish test result confidence. Therefore, this test method requires substantial personnel, time, and material, all of which result in significant cost expenditures in the evaluation of the load capacity of a lubricant.

1.2 Objectives

The objective of this program was to develop a lubricant load capacity tester that provides more consistent and repeatable test data results than those obtainable on the Ryder Gear Tester. The Twin Disc Gear Tooth Simulator developed in this effort uses an improved twin-disc concept to meet this goal.

Disc on disc type test machines have been evaluated in a number of attempts to simplify the evaluation procedure by using two rotating discs that can be loaded against each other instead of gears. The principal drawback with this type of machine is that, in any given test, it can evaluate only one slip percentage corresponding to one point on the profile of a gear tooth. A range of tests must then be run to evaluate lubricant performance as a function of slip.

The twin-disc rig concept investigated in this program for evaluating both lubricants and gear materials is one in which both discs are driven with cyclically varied speed (180° out of phase) to produce a range of repeatable slip velocities over one disc rotation, thereby simulating the variable slip that occurs in a gear mesh. The simpler disc geometry and test repeatability offered by this improved concept result in a significant reduction in the number of tests required for either development or specification test programs while still providing essential test variables.

In addition to providing data for ester- and hydrocarbon-based lubricants, the simulator also has the capability for testing fluorinated lubricants at temperatures to 700°F in an inert atmosphere. The Twin Disc Gear Tooth Simulator evaluated in the program described in this report is shown in Fig. 1.

1.3 Guide to Report

This report discusses the Ryder Gear Tester performance (Section 2.0), relates this performance to that of the Twin Disc Gear Tooth Simulator (Section 3.0), and describes the unique simulator design (Section 4.0). In addition, the test results for a variety of lubricants and several test disc geometric variations are presented to demonstrate both sensitivity and repeatability of the twin-disc design (Sections 5.0 and 6.0). Conclusions and recommendations are given in Section 7.0.

Supporting information is provided in the appendixes as follows. For reference purposes, Appendixes A and B include listings of parts lists and assembly drawings, respectively. Appendix C is the Hazard Analysis Report on the simulator and its subassemblies. Appendix D contains the Operating Instructions for the machine, and Appendix E contains the Maintenance Instructions.

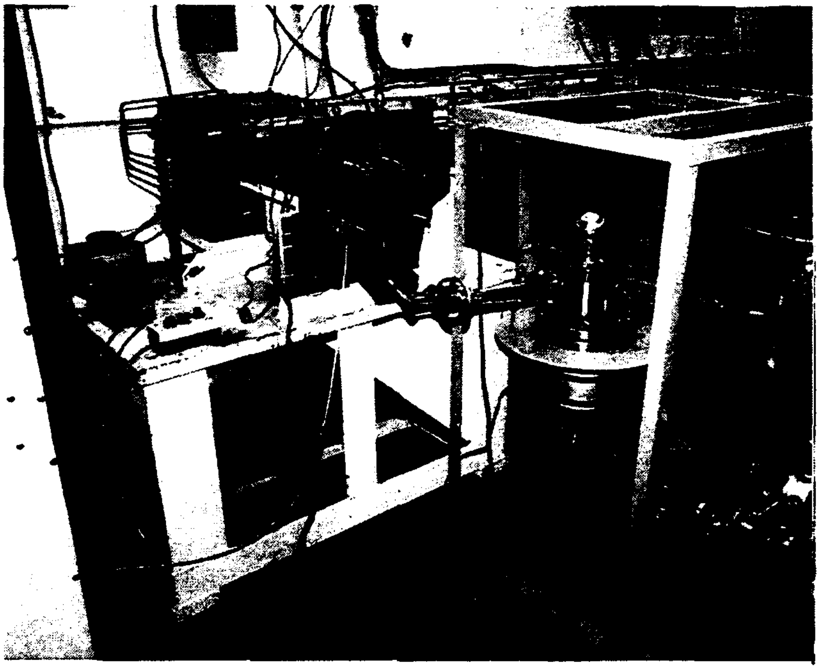

Figure 1 Photograph of Twin Disc Rig

2.0 RYDER GEAR PERFORMANCE PARAMETERS ANALYSIS

The development of any lubricant load capacity tester that would eventually replace the equipment used in current test methods must, as a minimum, attempt to duplicate the test parameters the present methods employ. The accepted method of evaluating lubricant load capacity makes use of the Ryder Gear Tester and follows the procedures described in ASTM-D-1947. This tester, which employs a set of special spur gears loaded against each other and rotated at a fixed speed, imposes a set of fixed operating conditions on the test gear teeth. The combined effect of Ryder gear design and tester operation provides the following parameters:

- Fixed rolling velocity
- Varying sliding velocity ("0" at the pitch line)
- Varying pressure-velocity (PV) product
- Varying critical flash temperature
- Maximum contact stress
- Minimum elastohydrodynamic (EHD) film thickness.

The following report sections present an analysis of the Ryder gear test that provides specific values for each of these test parameters.

For analysis purposes, the following Ryder gear test conditions were used:

- Rotational Speed: 10,000 rpm
- Lubricant Temperature: 165 ±5°F
- Maximum Test Load: 2960 lb/in.

The Ryder test gears are manufactured to the following geometry:

- Number of Teeth, N: 28
- Diametral Pitch, P: 8
- Pressure (Contact) Angle, ϕ: 22 1/2°
- Pitch Circle Radius, R_p: 1.7500 in.
- Outer Gear Radius, R_g: 1.8750 in.
- Base Circle Radius, R_b: 1.6168 in.

A contact diagram for the Ryder gear is shown in Fig. 2. With the use of this diagram, the tester's operating characteristics can be determined.

2.1 Rolling Velocity

Referring to Fig. 2, pure rolling occurs at the pitch circle with the tooth curvature radius BE, which is calculated as follows:

$$\begin{aligned}
\mathrm{BE} &= \sqrt{\left[R_p^2 + R_b^2 - 2\left(R_p R_b\right)\cos\phi\right]} \\
&= \sqrt{\left[1.7500^2 + 1.6168^2 - 2(1.7500 \times 1.6168)\cos 22\,1/2\right]} = \sqrt{0.4485} \\
&= 0.6697 \text{ in.}
\end{aligned}$$

At 10,000 rpm, with an angular velocity, ω(1047.2 radians/sec), the rolling velocity, V_r, is:

$$V_r = \omega\,\mathrm{BE} = 701 \text{ in./sec}$$

2.2 Sliding Velocity

To evaluate the Ryder gear sliding velocity, the driven gear and the driver gear starting angles (β and ξ, respectively) must be determined first. (See Fig. 2.) The angle at first contact for the driven gear rotating about center "A" is found by solving triangle BAC for β, which is found to be 7.92°. At the same gear positioning for the driver gear rotating about center "D," triangle BDC can be solved for ξ, which is found to be 8.59°.

For constant tester speed, the sliding velocity, V_S, during tooth mesh is given by:

$$V_s = \omega\left(R_2 - R_1\right)$$

where R_1 and R_2 are the instantaneous gear tooth radii at mesh.

Since

$$R_1 + R_2 = 2(0.6697)$$

then

$$V_s = \omega\left(1.339 - 2\,R_1\right)$$

For various angular positions through the gear mesh (varying ξ), R_1 is calculated from triangle DBE and the sliding velocity determined. Fig. 3 is a plot of the resulting ratio of the sliding velocity to the rolling velocity, V_s/V_r, as a function of the driver gear rotational position.

2.3 Pressure-Velocity Product

The maximum tooth load specified in ASTM-D-1947 is 2960 lb/in. for the Ryder gear with a width of 0.25 in. Assuming a 98% effective width, the tooth load becomes 755 lb.

The maximum tooth load does not exist for an entire tooth contact and varies as adjacent teeth pick up some of the load. The amount of individual tooth contact is determined by the contact ratio, M_p which is defined as:

$$M_p = \left\{ \left[N_D^2 \sin^2 \phi + 4(N_D + 1) \right]^{1/2} + \left[N_S^2 \sin^2 \phi + 4(N_S + 1) \right]^{1/2} \right\} / 2\pi \cos\phi$$

where

N_D = number of driver gear teeth
N_S = number of slave gear teeth

For the Ryder gears,

$N_D = N_S = 28$
$M_p = 1.54$

At M_p =1.54, a single tooth is in contact for 29.7% of available contact angle and the two teeth share load for the remaining 70.3%.

The total angle of contact on the driver gear extends from +8.59° to −7.92°. The load-sharing portion of contact is assumed as a linear load increase with full contact for [8.59 − (−7.92)](0.297) = 4.90°, so that the load curve would appear as shown in Fig. 4.

From this curve and the equations defining the sliding velocity, a relationship describing the velocity-load product PV can be generated. This relationship is shown graphically in Fig. 5. It is interesting to note that the maximum PV values occur near the extremes of the single tooth contact locations (at approximately the 4° position), and that at the pitch line where the sliding velocity is zero, or at the entering and leaving contact positions where the load is zero, the PV product is also zero.

2.4 Blok Critical Flash Temperature

The Blok [3] critical flash temperature, T_f, is defined by:

$$T_f = \left(0.062\, fW^{3/4} \left| \sqrt{V_1} - \sqrt{V_2} \right| E_r^{1/4}\right) / \beta R^{1/4}$$

where, for the Ryder gear machined from AMS 6260, the following parameters apply:[1]

f = friction coefficient (assumed to be 0.05)
W = unit load, lb / in. (tooth load divided by 0.25 divided by 98%)
V_1 = driver gear tooth tangential velocity variable, in./sec
V_2 = slave gear tooth tangential velocity variable, in./sec
E_r = elasticity factor $\approx E/(1-\nu^2)$
where
E = bulk mod ulus of gear material (30×10^6 lb / in.^{2})
ν = Poisson ratio of gear material (0.30)

[1] Values of these variables are the same as were calculated for the PV factor.

β = thermal contact coefficient = $\sqrt{C\rho k}$

where

C = gear material specific heat (1026 Btu / lb/° F)

r = gear material density (0.283 lb / in.^{3})

k = gear material thermal conductivity (5.61 Btu / sec ft^2 ° F / in.)

R = equivalent disk radius = $(R_1 R_2)/(R_1 + R_2)$

At these parameter values, the Blok critical flash temperature was calculated for one tooth mesh for the maximum load condition and normal operating speed (10,000 rpm) of the Ryder gear test. The results are shown in Fig. 6. A maximum Blok temperature, i.e. surface flash temperature above the bulk operating temperature, of approximately ΔT = 148°F occurs at the 5° mesh angle.

2.5 Maximum Contact Stress

Assuming perfect tooth alignment for the Ryder test gears, the rectangular Hertzian contact, σ_{Hz}, is given as:

$$\sigma_{Hz} = \left(\frac{W_t}{\pi f \cos\phi}\right)\left(\frac{\frac{1}{R_1}+\frac{1}{R_2}}{\frac{1-v_1^2}{E_1}+\frac{1-v_2^2}{E_2}}\right)$$

where, for the Ryder gear

W_t = unit load at the maximum test load (2960 lb / in.)

f = shape factor (1)

R_1 = radius at contact = R_2 (0.6697 in.)

v_1 = Poisson ratio = v_2 (0.30)

E_1 = elastic modulus = E_2 (30×10^6 lb / in.^{2})

At these parametric values, the calculated maximum Hertzian contact stress is:

$$\sigma_{Hz} = 2.15 \times 10^5 \text{ lb / in.}^2$$

2.6 Elastohydrodynamic Film Thickness

Computations for determining the Ryder gear EHD minimum film thickness were made using the MTI computer program HERTZPAN. The calculation showed that the minimum EHD film thickness, h, is 13.4×10^{-6} in.

2.7 Calculated Performance Parameter Summary

A summary of all the calculated parameters is presented below.

Average Roll Velocity: 701 in./sec
Maximum Sliding Velocity: 585 in./sec
Sliding Velocity at Maximum PV: 323 in./sec
Maximum External Load*: 755 lb
Maximum PV: 1.62×10^5 lb/in.-sec
Blok Temperature*: 148°F
Maximum σ_{Hz}*: 2.15×10^5 lb/in.^{2}
Minimum EHD film thickness h*: 13.4×10^{-6} in.

* Taken at the ASTM-D-1947 specification load of 2960 lb/in.

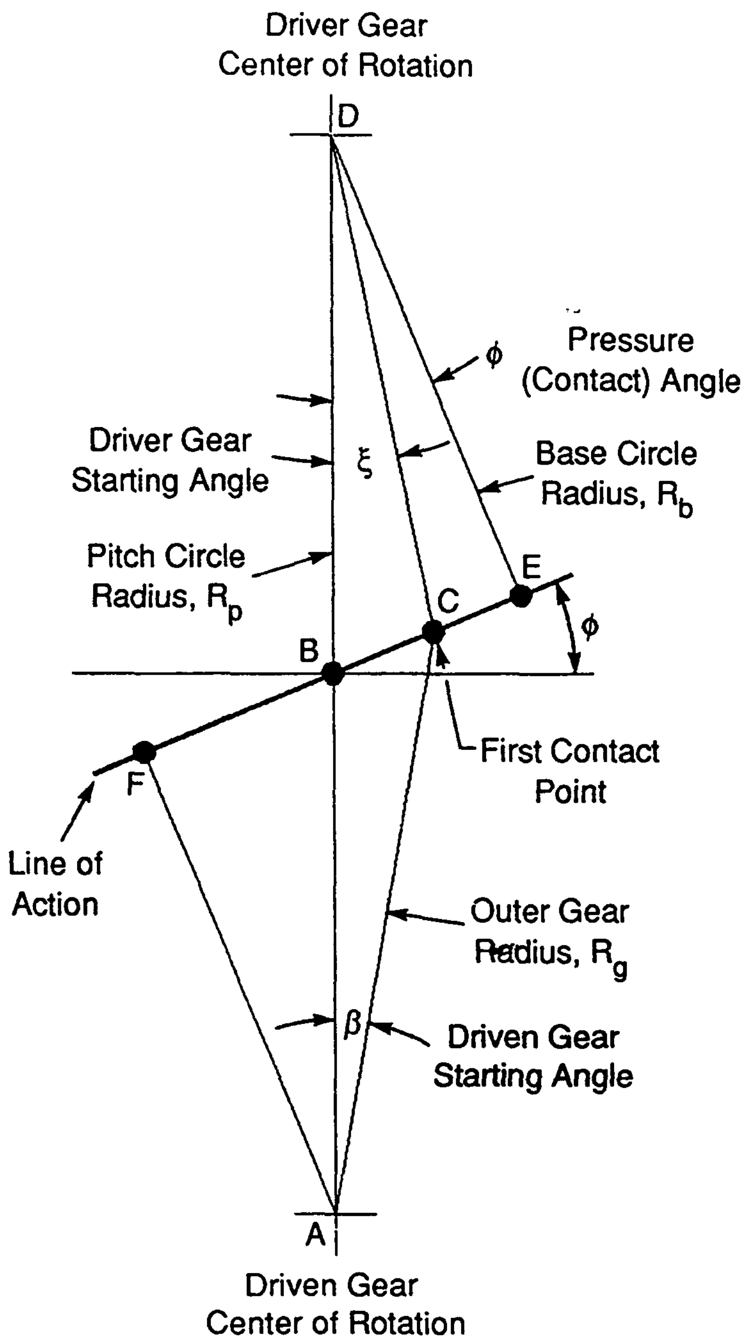

Figure 2 Contact Diagram for Ryder Gear

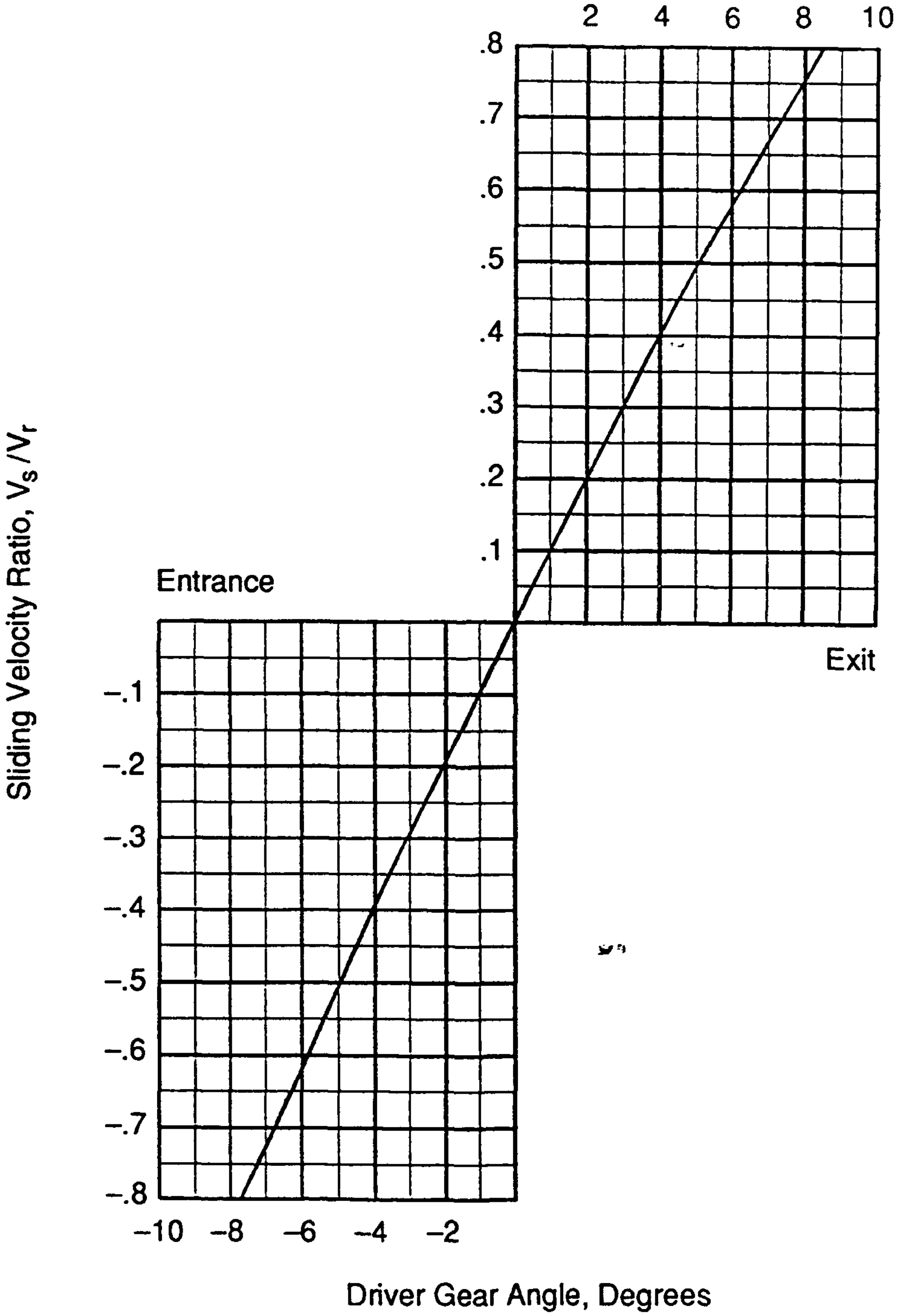

Figure 3 Sliding Velocity Profile for Ryder Gear Mesh

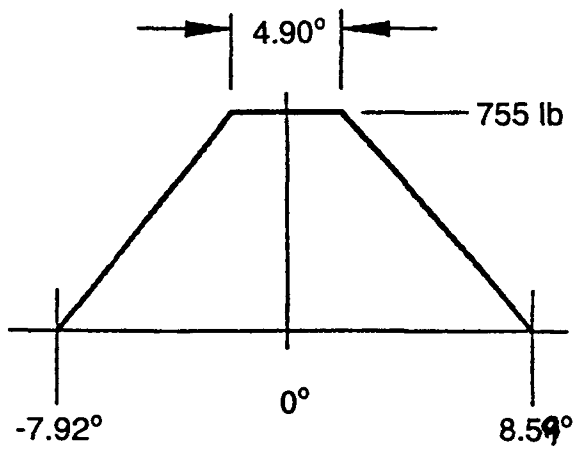

Figure 4. Tooth Load Profile for Ryder Gear

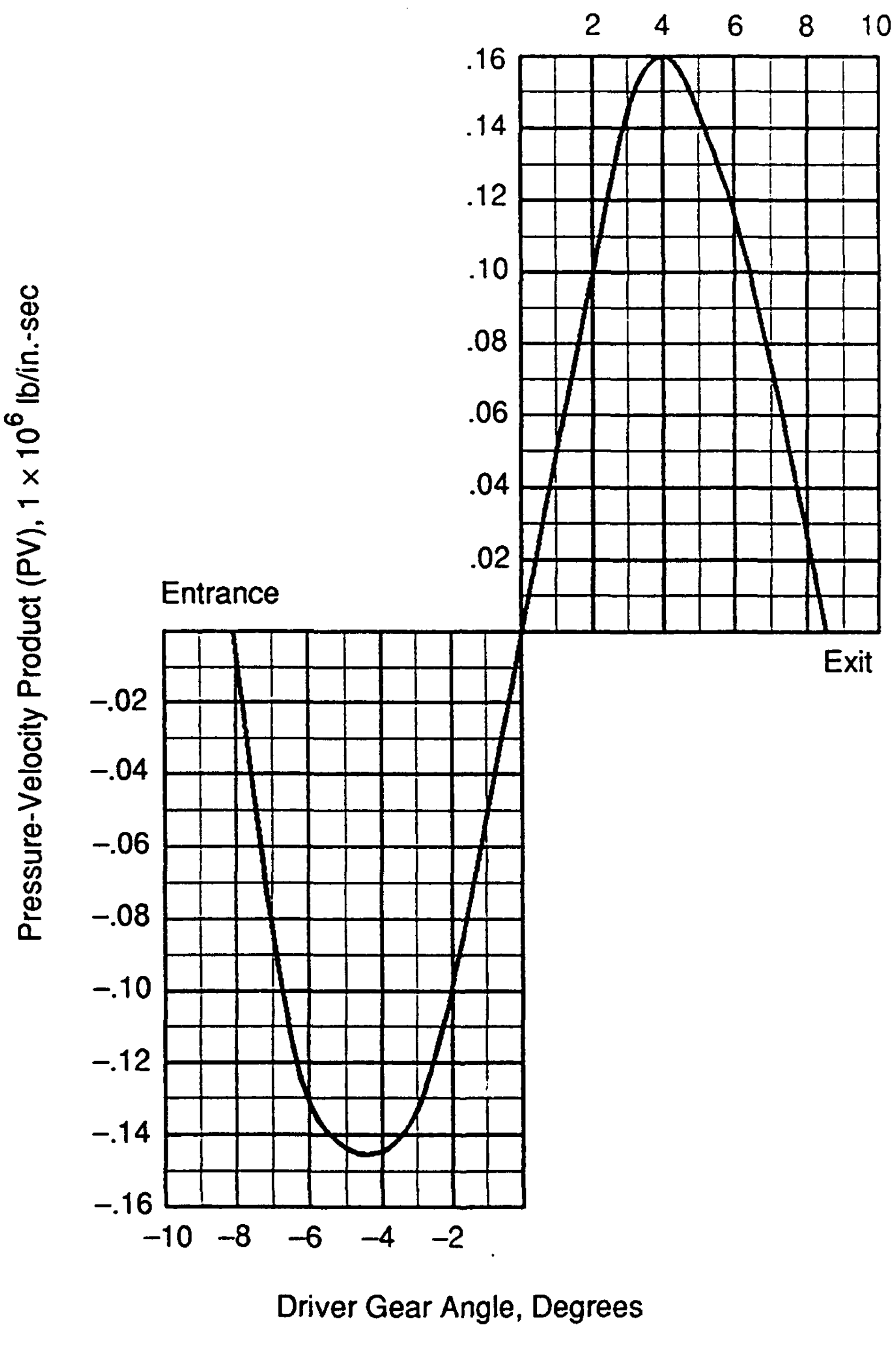

Figure 5 PV Product Versus Drive Gear Angle for Ryder Gear

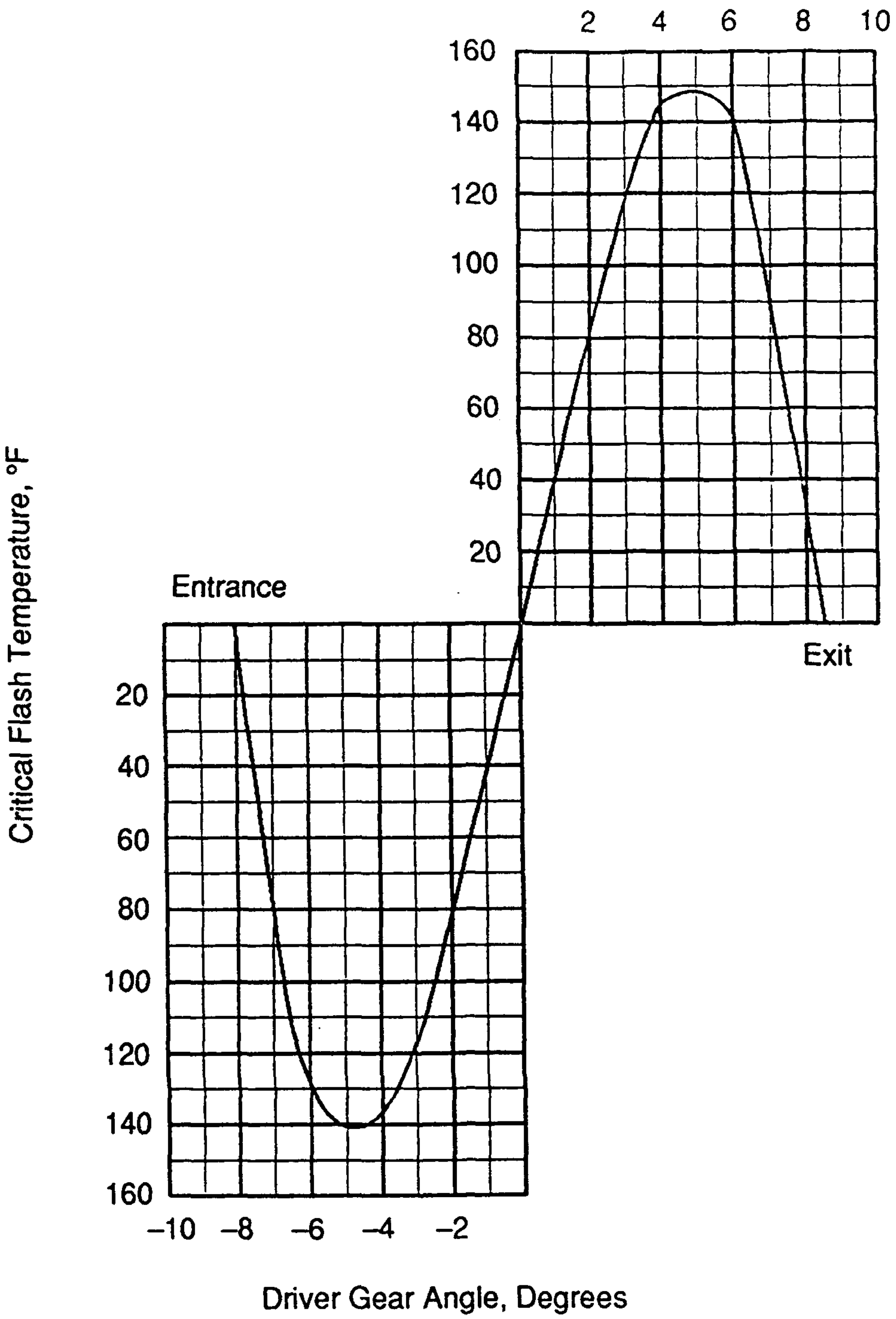

Figure 6 Critical Flash Temperature Versus Drive Gear Angle for Ryder Gear

3.0 TRANSLATION OF RYDER GEAR PERFORMANCE TO TWIN DISC SIMULATOR DESIGN

The Twin Disc Gear Tooth Simulator concept evolved from the desire to take the simple, fixed-slip rate, disc-type traction tester and incorporate controllable variable sliding velocities into its test profile. In this way, the reduced cost and more easily controllable quality of the test components (discs as opposed to special gears) could be utilized to reduce the variability shown by the Ryder gear test method. At the same time, the problems found in using disc tests at fixed slip conditions to evaluate load capacity could also be avoided. If a controlled, variable-slip rate could be superimposed on existing parameters such as the adjustable rolling velocity and load already available on disc-type machines, then many of the more critical test parameters of the Ryder Gear Tester would be simulated.

3.1 Establishing Simulator Parameters

The criteria used in establishing the basic simulator parameters were to duplicate as closely as possible the lubricant test conditions found in the Ryder gear test.

The nominal speed and, therefore, the roll velocity parameter for the simulator is a direct result of the test disc diameter selection. The disc diameter, in turn, is dependent upon the allowable applied load for the disc support system and the size of the required drive train components. A convenient disc size of 4 in. was selected to begin the system conceptual design.

The 4-in. disc diameter quickly established two parameters. The first, derived from the Ryder Gear Tester average roll velocity of 701 in./sec, was the nominal simulator rotational speed, which became 3347 rpm. The second parameter set by the test discs was the external load necessary to obtain a 200,000 lb/in.^{2} Hertzian stress. This level of Hertzian stress was best achieved by choosing a fairly large crown radius; at an 18-in. radius, a contact load of 1000 lb achieved the acceptable Hertzian stress of 203,000 lb/in.^{2}. A broad crown radius was preferred in the disc geometry since it would make the results less sensitive to small errors in crown radius in the manufacturing process than with a smaller crown radius. The same absolute machining error in crown radius results in a greater error in absolute radius at small crown values which produces more variability in the stress level in the contact with the smaller crown. A disc with smaller crown radius would also subject the contact to a greater variation in stress for a given machine dynamic load. With the basic disc geometry defined, the next design step was to settle on a way to drive the discs to achieve the necessary variable slip velocity.

3.2 Establishing the Disc Drive System

At first thought, the system needed to drive the two test discs could consist of two interdependent drives with controlled, variable-speed capability. The drive speed controller could then be programmed to provide variable slip velocities. This type of disc machine has already been built, but it does have drawbacks such as:

- Extensive drive system and speed controls to vary both roll and slip velocities

- Extreme speed control precision to maintain the relationship between specific PV locations on each disc. This feature is important since in the Ryder gear test, the same two teeth always mesh on the driver and driven gears. This has been shown to be important in scuffing tests because damage in the contact tends to propagate due to the repeated contact of the same points once damage has occurred.

- Large drive power requirements.

A way to provide both a fixed roll velocity and a controlled and a variable slip velocity without the need for expensive drives and controls is with a unique transmission employing non circular gears. With this type of transmission coupled to a pair of ball bearing spindles, a Twin Disc Gear Tooth Simulator with controlled variable slip and constant (but adjustable) roll velocity could be constructed. The basic simulator configuration is shown in Fig. 7.

3.3 The Variable Slip Transmission Concept

The heart of the simulator is its transmission, which evolved through a number of developmental stages until its final version emerged.

Initially, the transmission design took the form shown in Fig. 8. In this design, two output shafts, although driven by a common motor, had widely varying speed profiles. One shaft was driven via a toothed belt at constant speed. The second shaft was driven via an identical toothed belt, but with a non circular gear set included in the power train. The introduction of the non circular gears provided the proper rotational direction for both disks along with the required repetitive varying rotational speed and reversing slip velocity. (Two non circular gear types are elliptical and sinusoidal [4].) At this point, however, some very difficult design problems arose. These problems are summarized below:

- With one disc operating at a fixed speed and a second operating at a variable speed, the average roll velocity $(V_1 + V_2)/2$ was not a constant. Changes in average rolling velocity will change the EHD film thickness around the disc, a factor felt to be undesirable in this design.

- The use of an elliptical gear set generated very high acceleration rates. For a two-lobe gear set with a maximum ratio of K = 2, the angular acceleration approached two times the square of the angular velocity of the fixed speed gear. At a speed of 3350 rpm, the angular acceleration became $2.5 \times 10^5/\text{sec}^2$. Even with a modest drive train inertia of 30 lb-in.^2, drive torques would approach 1700 ft-lb, a torque much too high for any practical design.

- Switching the drive line acceleration from positive to negative while driving the system with a tooth belt arrangement left serious questions about belt durability and torsional stability.

- Reflected torque was high since the variable torque requirement was unbalanced.

It became obvious that a concept shift was necessary and the following changes were examined and implemented:

- Varying the speed of both discs so that neither disc has to make the total angular velocity change

- Limiting the maximum disc slip velocity to the Ryder gear slip velocity at its maximum PV point

- Changing from elliptical to sinusoidal gears [4] to produce the same disc sliding speed changes with lower acceleration rates. A sinusoidal gear set is one in which a constant angular velocity on the driver gear produces a sinusoidal angular velocity change on the driven gear symmetrically centered about the mean angular velocity.

The transmission configuration resulting from this rethinking is shown in Fig. 9.

At a nominal speed of 3356 rpm, a speed variation of only ±725 rpm provided an acceptable slip velocity of 304 in./sec (for a 4-in. diameter disc) and a PV product of 3.04×10^5 lb/in.-sec at a 1000-lb load. In this arrangement, the reflected torque was exactly balanced when the right-side sinusoidal gears were phased at 180° from the left-side gear set. At a phasing of 180°, the decelerating torque on one drive train is always equal to the accelerating torque on the other. Therefore, the driving gears are never unloaded and the drive motor must supply only system losses.

A serious drawback, however, existed with the drive train as shown in Fig. 9. With this configuration, the change from acceleration to deceleration caused the take-up of backlash in the sinusoidal gears, which at high torque levels, could induce gear damage and eventual tooth fatigue.

The solution to the backlash problem was to build each disc drive line as an independent four-square system with a means for torquing the lines to eliminate backlash. Fig. 10 shows the double four-square configuration. In this arrangement, two identical sinusoidal gear sets are used. The driving sinusoidal gears are mounted along with conventional spur driver gears on each lower shaft; these shafts are actually a two-piece design with a precisely positioned axial dowel aligning the two sections. The dowel maintains alignment of the shaft sections and is never removed. A bolted flange rigidly connects the shaft sections, the flange bolts are loosened, and the shaft sections torqued to remove any backlash at the sinusoidal gear sets.

A light torque of only 10 to 15 ft-lb is needed to maintain tooth contact since the windup of the lighter upper shaft is negligible. In addition, the twisting torque can be applied in any direction since the acceleration and deceleration torques will always act against the loaded side of the sinusoidal gear teeth.

The selection of the disc drive transmission concept permitted the finalization of the Twin Disc Gear Tooth Simulator design.

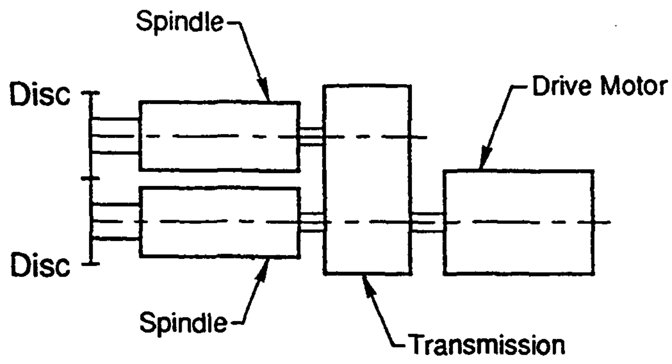

Figure 7 Basic Twin Disc Simulator Configuration

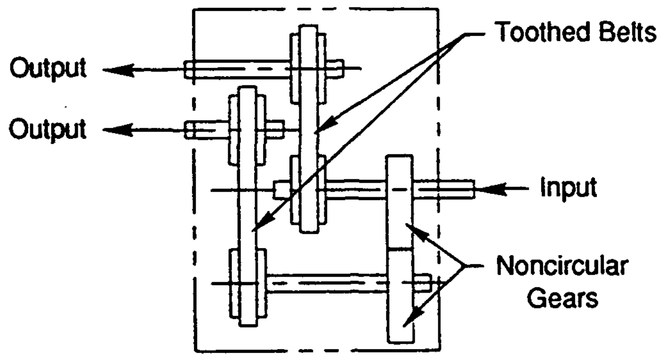

Figure 8 Initial Transmission Design

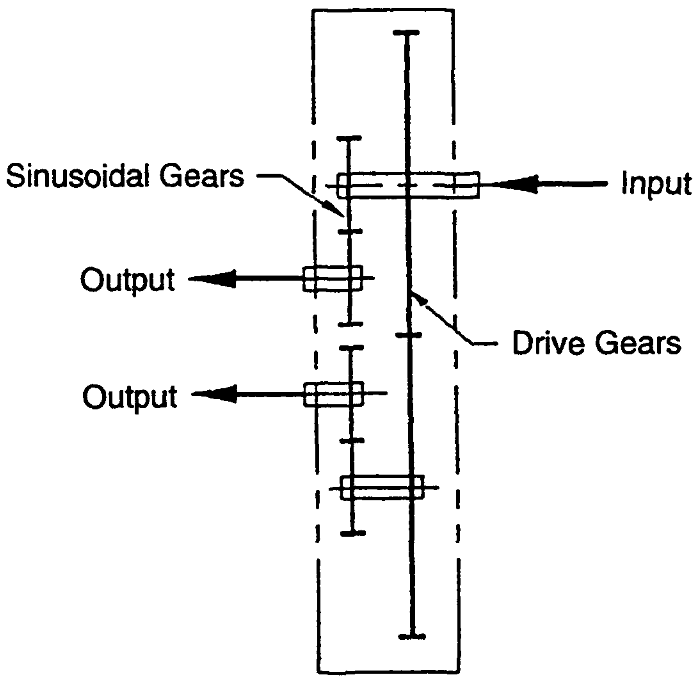

Figure 9 Initial Transmission Design with Sinusoidal Gears

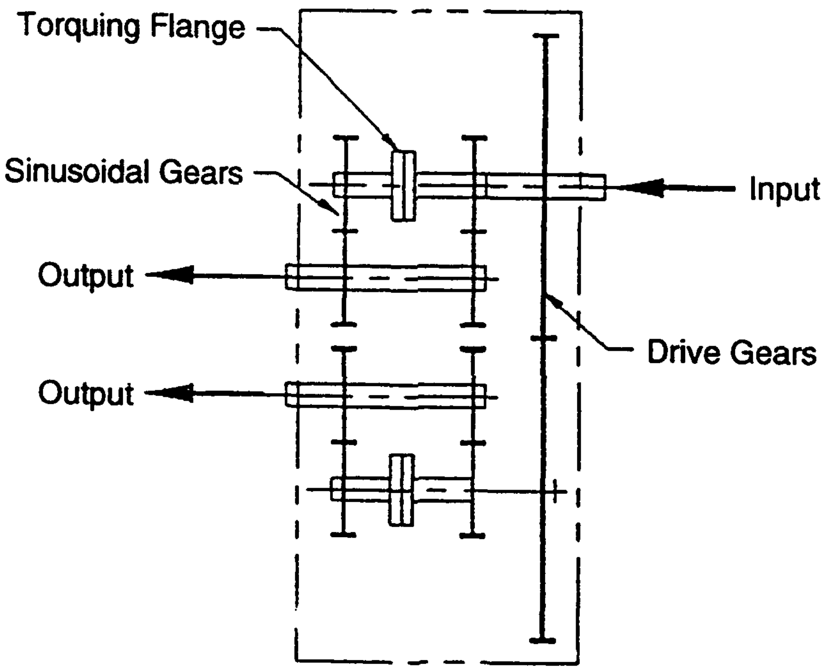

Figure 10 Zero Backlash Final Transmission Design

4.0 FINAL TWIN DISC GEAR TOOTH SIMULATOR DESIGN

The design process to develop a viable simulator configuration, based on sinusoidal gears to provide a variable-slip velocity profile, proceeded along the following path:

1. Complete the transmission kinematic design to establish both maximum slip velocities and acceleration rates

2. Size drive line components (discs, disc mounts, spindles, and couplings) and establish system inertia and parasitic losses from seals, bearings, and windage.

3. Based on the driven inertias established in Step 2, design transmission gearing and shafting and finalize the transmission packaging.

4. Configure the drive line components into a viable system.

5. Design the simulator support structure and ancillary systems for disc and transmission lubrication.

The report sections that follow describe the results of these design steps. The figures included in these sections represent the general concepts involved in the simulator design. The size and number of the final detailed design drawings precludes their use in this report. A complete listing of these drawings is included in Appendix B for reference.

4.1 Sinusoidal Gear Kinematic Design

The sinusoidal gear kinematic design was based on the following requirements:

- The sinusoidal gears that drive the upper transmission shaft had to clear each other at the 4-in. shaft center distance established by the test disc diameter.

- The speed variation necessary to achieve a maximum slip velocity at a nominal rolling velocity of 701 in./sec was to be equally divided between the two discs.

- Gear width would be established after the system inertia was determined, the peak acceleration torque calculated, and the parasitic losses established.

The maximum speed variation that can be obtained by the sinusoidal gears in the Twin Disc transmission is governed by the center distance of the two output shafts. Even though the sinusoidal gear set driving one disc may operate on a center distance of less than 4 in., the relationship between the two gear sets driving a pair of discs puts the maximum gear radii in juxtaposition to each other. This arrangement limits the radial offset of the individual sinusoidal gears.

The equations describing sinusoidal gears as reported by Cunningham [4], are:

$$\theta_{out} = (C_1 + C_2)\,\theta_{in} - \left[(57.29\,C_2)/(C_3)\right]\sin(C_3\,\theta_{in})$$

$$\omega_{out}/\omega_{in} = C_1 + C_2\left[1 - \cos(C_3\,\theta_{in})\right]$$

$$\alpha_{out} = C_2\,C_3\,\omega_{in}^2\,\sin(C_3\,\theta_{in})$$

where:

θ_{out} = input angle, degrees
C_1 = ratio at start of transition (at $\theta_{in} = 0$)
$C_1 + C_2$ = ratio at end of transition (at $\theta_{in} = 0$)
C_2 = 1/2 (ratio at end of transition - ratio at start of transition)
C_3 = scale factor such that $C_3 \times \theta = 180°$
ω = angular velocity, sec^{-1}
α = angular acceleration, sec^{-2}

Given C_1, the ratio change, RC, for pure sinusoidal gears is:

$RC = (2 - C_1/C_1)$

For the Twin Disc transmission, one revolution on the input gear will produce one revolution of the output gear, thus making $C_3 = 1$.

The remaining constants, C_1 and C_2, for sinusoidal gears can be determined from the required speed variations that are derived from the Ryder gear calculations. The slip rate for the Ryder gear at its maximum PV is about 304 in./sec. If this slip velocity is obtained by changing the speed of both discs equally (to maintain a constant average roll velocity), then the required velocity change at each disc is 152 in./sec. On a 4-in. diameter, this linear velocity change translates to an angular velocity change of 76 sec^{-1} based on a nominal angular velocity of 350 sec^{-1} (to provide the desired average roll velocity). The required variation of angular velocity for the transmission output shafts is:

$\omega_{in} = 350\ sec^{-1}$
$\omega_{out\text{-}min} = 274\ sec^{-1}$
$\omega_{out\text{-}max} = 426\ sec^{-1}$

Thus, at these rates:

$C_1 = 0.78286$
$C_2 = 0.21714$

For these values, the maximum angular acceleration rate, α, produced by the gear set is:

$\alpha = 2.66 \times 10^4\ sec^{-2}$

This value is significantly less than the higher rate of $\alpha = 2.6 \times 10^5\ sec^{-2}$ calculated for the elliptical gears initially evaluated for the transmission.

The final kinematic design for the transmission permitted the complete evaluation of the disc test parameters, which are listed below:

Average Roll Velocity: 701 in./sec
Maximum Slip Velocity: 304 in./sec
Nominal Load: 100 lb
Maximum PV (at 1000-lb load): 3.04×10^5 lb-in./sec
Test Disc Hertzian Stress (at 1000 lb and 18-in. crown radius): 2.03×10^5 lb/in.^{2}
EHD Film Thickness: 1.88×10^{-5} in.
Block Critical Flash Temperature: 283°F

4.2 Drive Line Components

The design of the drive line components, which include the test discs, heat dams, and spindles, was based on the need to provide a minimum-inertia system that would limit the transmission gear tooth loads while maintaining safe stress levels. To accomplish this goal, several specific design features were insisted upon:

- All joints between components that are to be disassembled only for repair purposes were made with shrink fit, three-lobe polygons.
- All joints between components that were to be disassembled infrequently were plotted with three-lobe polygons to absorb torque and were flange-bolted together.
- Test discs were to be held in place with sliding, nonlocking, nested, tapered rings* and piloted with a light shrink fit on the disc mount.

With these criteria in mind, the tester design was produced, with the following results.

4.2.1 Test Disc

From previously established design criteria and testing goals, the basic test disc geometry evolved to the configuration illustrated in Fig. 11. The inertia moment for this configuration is 4.67 lb-in.^{2} and, therefore, requires a torque input of 322 in.-lb to be accelerated by the Twin Disc transmission. For an assumed maximum traction coefficient of 0.05 and a radial load of 1000 lb, friction driving torque is 100 in.-lb. The net torque required at the disc mount for proper acceleration is 422 in.-lb.

Ringfeder-type locking elements with a torque transmission capability of 1900 in.-lb were selected to drive the test discs. The locking element clamping discs add 3 in.-lb to the driving torque requirement, but are directly coupled to the drive train and are not felt by locking elements.

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* Ringfeder Locking Elements - Ringfeder Corp., Westwood, New Jersey.

4.2.2 Heat Dam

A heat dam was designed to protect the spindle bearings from the expected high temperatures at the test discs. The heat dam, inserted between the test discs and the spindle shaft, provides the necessary restriction to heat flow to protect the spindle bearings. The heat dam establishes a thermal gradient of sufficient magnitude to permit safe spindle bearing temperatures at disc temperatures as high as 800°F.

The final heat dam geometry is illustrated in Fig. 12. Its polar inertia moment is 2.35 lb-in.^{2} and requires a peak acceleration torque of 162 in.-lb. The total load to be transmitted through the heat dam, which includes the disc inertia and friction load, is 585 in.-lb.

The heat dam is capable of safely sustaining the combined effect of cyclic torsional stress and cyclic bending stresses generated by the radial disc load. To confirm the excess capacity (safety factor), the heat dam's highest calculated stress resulting from a disc load of 2000 lb (twice the design value) was found to be below the endurance limit for the heat dam material.

4.2.3 Spindle

A common spindle design was established to support each disc/heat dam assembly. the major design premise was that the spindles should stand alone without the need for an auxiliary lubrication support system. Included in the design guidelines was a minimum inertia system with reasonable life.

The final spindle configuration is shown in Fig. 13. The ball bearings are a deep-groove Conrad design mounted in pairs with outside shields. The bearings have steel ribbon retainers and are lubricated with Mobil SHC 32 grease, which permits sustained bearing operating temperatures to 250°F. The L_{10} life of the four-bearing set for each spindle is 4300 hr at a disc load factor of 1.25 times the design load of 1000 lb.

The driving polygon at the heat dam is sized for a minimum torque of 694 in.-lb. The bolted flange is designed to restrict the maximum bolt loading of six-bolt pattern to 600 lb. This external load limit permits the use of 1/4-20 grade 8 bolts torqued to 9 ft-lb (providing an at-rest clamping load of 2850 lb/bolt).

The polar inertia moment for the rotating components of the spindle is 6.59 lb-in.^{2} and requires an acceleration torque of 454 in.-lb with a total system torque load of 1039 in.-lb. A polygon mount connects the spindle to the drive coupling. The 1-in. polygon selected for this torque level has a shear safety factor of 3.

4.2.4 Drive Coupling

The connections between the spindle input shafts and the transmission output shafts were made using modified commercial double-fitting disc couplings. The modifications were made to reduce the coupling's inertia moment value and providing polygon mounts instead of the normal cylindrical shaft fits and keyways. The selected couplings are Rexnord series 53, size 125, modified to produce an inertia moment of 5.08 lb-in.^{2} and torsional stiffness of 3.45×10^5 in.-lb/rad.

The input torque requirement at the coupling is the sum of all the connected products of the inertia and angular accelerations in the drive train plus the contact friction torque. The total driven inertia is 18.74 lb-in.^{2} for a total inertia torque of 1289 in.-lb and a net torque of 1390 in.-lb to be transmitted through the coupling. The selected coupling's safety factor at full load is 1.3, based on its commercial capacity, which is already factored for safety. The polygon sized for the input side of the couplings is 1.125 in. with a safety factor of 3.

4.3 Transmission

The kinematic design for the sinusoidal gearing was established in Section 4.1. The final design of the remaining gear parameters is described in this section. Following the gearing determination and after the remainder of the transmission components such as bearings and seals were selected, the transmission configuration was established and the housing designed.

4.3.1 Sinusoidal Gear Sizing

Each sinusoidal gear is designed for a speed of 3400 rpm. The operating center distance of the gears was arrived at by limiting the pitch circle radius plus gear tooth addendum sum for two gears operating side-by-side to 4.00 in. (the test disc center distance). An additional limitation required that the sinusoidal gear sets (driver and driven) not only have the same number of teeth but also that the number be odd. The final gear design produced the remaining gear data:

Number of Teeth: 37
Diametral Pitch: 10
Operating Center Distance: 3.750 in.

The face width of the sinusoidal gears was sized to sustain not only the acceleration torques from the external drive line components, but also the acceleration torques generated by the driven sinusoidal gears, the upper transmission shafting, and the test disc friction.

The calculated inertia moment for each sinusoidal gear is 7.77 lb-in.^{2}; each upper shaft inertia moment is 4.09 lb-in.^{2}. The total inertia and the resulting peak torque to be driven by each disk drive line is listed below.

Component	Inertia (in.-lb^2)	Peak Torque (in.-lb)
Disc	4.67	322
Ringfeder	0.05	3
Heat Dam	2.35	162
Spindle	6.59	454
Coupling	5.08	350
Gears (2)	15.54	1071
Shaft	4.09	282
Friction	--	100
Total	38.82	2722

A face width of 1.375 in., including a generous 1/16 in. tooth rounding radius, proved adequate for handling the loading.

To provide superior strength, the sinusoidal gears were manufactured from a case-hardenable, chrome-vanadium steel identified by the German designation DIN 31 CR MO V9. The four pairs of sinusoidal gears in the transmission are mounted with an interference polygon fit to eliminate the need for hubs and stress-raising keyways. The gear mounting polygons were sized large enough to fit over the transmission shafting bearings seats but small enough to minimize the bore stress effects on the sinusoidal gears. The resulting polygon selection was:

Upper Shafting: 1.500 in. P3 polygon, fit Class F-3
Lower Shafting: 1.750 in. P3 polygon, fit Class F-3

Before the placement of transmission gear shafting could be finalized, the power input drive gear pair had to be configured. The driving gears could not be designed until the total power requirement of the transmission was determined, which, in turn required calculating the bearing and seal losses.

4.3.2 Bearing and Seal Sizing

The bearings selected for supporting the transmission shafting were sized to support the radial loads resulting from the maximum driving torque at the sinusoidal gears. Simple lip seals are used at the one input and two output shaft locations. The final bearing selection is given below:

- Lower Transmission Shafts
  - 108 RDS-DB, Duplex Ball Bearing
  - HJ-283780*, Roller Bearing
  - IR-222820*, Roller Bearing

- Upper Transmission Shafts
  - HJ-324120*, Roller Bearing
  - IR-263220*, Roller Bearing

4.3.3 Power Loss Calculations

The component parts of the transmission power losses calculated at 3400 rpm are summarized below:

- Bearing Friction
  - Total for four ball bearings: 0.25 ft-lb
  - Total for six roller bearings: 0.67 ft-lb

- Seal Friction
  - Total for three seals: 1.27 ft-lb

* Torrington Bearing designation.

- Windage: 2.40 ft-lb

- Gear Friction
  - Total for five meshes at 0.47 ft-lb: 2.35 ft-lb

In addition to internal losses, the transmission power train must supply three additional external losses, which are:

- Disc Friction: 100 in.-lb

- Spindle Bearing Friction
  - Total for eight bearings at 0.20 ft-lb: 1.6 ft-lb

- Windage: 0.5 ft-lb

The transmission's internal power loss is 4.2 hp at 3400 rpm. External losses total 6.6 hp, resulting in a total calculated system loss of 10.8 hp.

4.3.4 Driving Gear

The input gear set supplying the power to drive the Twin Disc simulator was established as a spur gear design operating with a one-to-one ratio at 3400 rpm. The gear set was designed for high durability by setting a minimum transmitted power level of 50 hp, much higher than the gears would ever experience in practice. The resulting gear design is identified below:

Number of Teeth: 61

Pitch Diameter: 6.100 in.

Outside Diameter: 6.300 in.

Face Width: 1.000 in.

Base Circle Diameter: 5.7321 in.

Operating Center Distance: 6.100 in.

Operating Pressure Angle: 20°

Contact Stress: 77,646 psi
  Safety Factor: 1.78 at 50 hp

Bending Stress: 12,890 psi (max)
  Safety Factor: 4.10 at 50 hp

Material (Case-Carburized Steel)
  Case Hardness: 50 Re
  Core Hardness: 314 BHD

4.3.5 Transmission Configuration

After the driving gears were sized, the final configuration of the transmission was established. To provide a more compact design, the driving gears were set between the sinusoidal gears in each four-square set alongside the torquing flanges on the lower shafting. Fig. 14 illustrates the relative position of the transmission shaft center lines. The configuration of each four-square subassembly is illustrated in Fig. 15. A perspective view of all four transmission shafts is shown in Fig. 16.

4.3.6 Transmission Lubrication

An independent, self-contained lubrication system supplies the transmission. This system is shown schematically in Fig. 17. No external cooler is employed; sufficient heat loss through the reservoir maintains a safe oil temperature. Oil jet lubrication is used for all the gear meshes and the ball bearings; central feed manifolds are used to lubricate the roller bearings. The net bearing and seal power loss of 1.4 hp dictated a Mobil SHC-630 lubricant flow rate of 2.5 gpm proportioned according to loss. The net calculated temperature rise is only 58°F. Orifice fittings at each bearing location apportioned the appropriate oil flow rate at a delivery pressure of 35 psi dictated by the gear lubrication requirements. Each gear mesh jet provides a fan shaped oil spray and delivers 0.56 gpm at the manifold pressure of 35 psi.

4.4 Drive Line Component Integration

The viability of the Twin Disc simulator depends in some part on configuring its many components into an easily used entity. These components, exclusive of the test lubricant supply system, are combined into one assembly supported by a substantial welded machine frame. To accomplish the integration, the methods described in the following subsections were employed.

4.4.1 Spindle Mounting

The ability to load one test disc against another dictates the necessity of relative motion between the two spindles that support the discs. The design constraint imposed on the movement of one spindle relative to the other is that the couplings connecting the spindles to the transmission can have no lateral misalignment. This constraint was obviated by the following configuration.

One of the two spindles is firmly attached to the machine base. The second spindle, called the movable spindle, is mounted on a pivoting arm that is free to articulate about a vertical center line passing through the midspan of the spindle to a transmission drive coupling. With no external load applied, the two spindle shafts are parallel with a gap of 0.06 in. between the test discs. When loaded horizontally, the movable spindle rotates about the pivoting arm axis and loads the test discs against each other. The misalignment angle at the drive coupling is well within coupling capability.

4.4.2 Test Disc Enclosure

The test disc enclosure, referred to as the test head, is designed as a modular subassembly that can be easily removed from the test stand. The test head assembly is shown in Fig. 18. It houses the test discs, their respective spindle shaft seals, the test lubricant inlet jet, and the disc heating elements. These elements are electrical resistance heaters that aid in maintaining internal temperatures. Fig. 19 shows the installation of these heaters; Fig. 20 shows their electrical connections.

Two covers are included with the test head: a high-temperature plastic one suitable for lubricant temperatures to 170°F and a metal one for elevated-temperature testing. The entire test head is fabricated from Inconel 718 to permit testing with lubricants that may not be compatible with ordinary steels.

To minimize seal cost while preserving the ability to tolerate the lateral movement of the movable spindle, externally pressurized (hydrostatic) ring seals are used rather than more expensive face seals. The seals are pressurized to between 35 and 40 lb/in.^{2} and fully contain the test lubricant within the test head. An added feature of the hydrostatic ring seal becomes evident when an inert atmosphere is required with the test head. The seals can be pressurized with any gas required; the gas leakage through the seals into the test head would act as a cover gas.

4.4.3 Transmission Mounting

The transmission case contains a mounting flange on its lower housing. The complete transmission is bolted directly to the machine base. The two output shafts are connected to the support spindles via the couplings described in Subsection 4.2.4.

4.4.4 Motor Selection

Power loss calculations indicated that a total power requirement of 10.8 hp would be necessary to drive the simulator. To satisfy this need with ample margin for error, a 25-hp, 3600-rpm motor was selected. Variable-speed capability is provided by a solid state inverter (Laser Model 2950-8402). A toothed belt suitable for the drive motor was selected to bring power from the motor to the transmission's input shaft. Fig. 21 shows the connection schematic of the motor controller for three-wire control and the connection for the transmission oil pressure interlock.

4.5 Ancillary Systems

Two ancillary systems are incorporated in the Twin Disc Gear Tooth Simulator design. Their designs are described as follows.

4.5.1 Test Lubricant Supply System

A separate and independent supply system, shown schematically in Fig. 22, was designed to handle almost all possible test lubricants to a temperature of 700°F. Lubricant is supplied at a slight gravity head to a canned rotor centrifugal pump through a manual shutoff valve and a sintered stainless steel filter. The shutoff valve is used to stop the lubricant supply to allow access to the lubricant pump for maintenance. The combination of a bypass and trim valve is used to adjust test flow while allowing sufficient flow through the pump to prevent excess churning losses.

For test lubricant heating, a combination of sump heaters and line heaters are used. The electrical connection for the sump heater is shown in Fig. 23. The electrical connection for the supply line tape heater is shown in Fig. 24.

4.5.2 Loading System

A combined pneumatic/hydraulic loading system, shown in Fig. 25, is used to apply test loads in the following manner. Shop air controlled by a pressure regulator acts against the large-diameter piston of the hydraulic booster. Hydraulic oil, now at a proportionally higher pressure at the hydraulic booster's small-diameter piston, activates the load cylinder, which then pushes against the movable spindle through a load transducer.

Since the load application point is closer to the movable spindle hinge point (9.625 in.) than is the contact point on the test discs (19.38 in.), the actual disc load is smaller than the transducer load. For the simulator, as presently constructed, the proportionality constant between the test and applied loads (i.e., Test Load/Applied Load) is 0.626.

To remove the test load, a four-way valve is used to switch the regulated air pressure from the hydraulic booster to the unloading side of the load cylinder.

4.6 Instrumentation

While not extensive, instrumentation is critical to the operation of the Twin Disc rig. For machine control and health monitoring, a speed pickup is provided on the motor shaft and thermocouples have been mounted in the critical drive spindle bearings, the gear box oil sump and in the gearbox oil jets. For load measurement and control, a load cell is mounted on one of the spindle housings and a load cylinder is used to apply the load. Accelerometers have also been placed on the spindle housings and the gearbox to monitor vibration. These temperature, load and speed conditions insure that the desired test conditions are maintained during a test, and provide warning of any problem which may be developing in the critical mechanical components such as bearings and gears.

In addition to the machine performance related instrumentation, there are also test lubricant thermocouples located in the lubricant sump, in the supply lines and in the jet nozzle to measure temperature just prior to jetting the lubricant into the disc contact. The disc support spindles were also isolated from ground and the rig was instrumented with an instrument for measuring the contact resistance across the lubricant film. This contact resistance measurement was included in the rig design in the hope that it could be used for a measurement of when the lubricant load capacity had been reached based on a drop in contact resistance as the film disappeared. The hope was that this would provide a more sensitive and easier measure of the loss of the lubricant film than the standard process of running a rig at a given load for a fixed time, stopping and inspecting the test samples for evidence of scuffing. During the course of the test program unfortunately it became evident that contact resistance was not sensitive enough to be used as a determination of load capacity. What was found was that the technique worked moderately well for unformulated lubricants with scuffing failure occurring relatively close to the load at which the lubricant film could be determined to disappear as measured by contact resistance. Unfortunately, for fully formulated lubricants it was found that even after contact resistance became essentially zero (as might be expected when the discs are operating in the severe boundary regime in the high slip region) the load on the discs had to be increased significantly before scuffing would occur. Since the technique could not be used to determine scuffing load, it was not used directly in the evaluation of scuffing load once these results were found. The contact resistance measurements are still used to give the operator an indication of the starting point for step loading evaluations designed to visually find scuffing. In this way slowly stepping through lower loads where there is no chance of scuffing can be eliminated and the test cycle shortened.

It seems that if more time could be spent to investigate the details of how the contact resistance changes with load and additive content the difference between the load at which the film thickness becomes minimal and that at which scuffing occurs could be of significance in evaluating lubricant formulations. It was not possible to evaluate such effects as part of this effort. One interesting aspect of the contact resistance measurements was the fact that they confirmed the effects of the varying rolling and sliding motion on the film thickness as the discs rotated. The film thickness first becomes zero in the regions of highest sliding with the thickness increasing toward the pure rolling positions. As the load increases above that at which the film disappears in the high sliding region, the region of zero contact resistance spreads until in some cases contact resistance indicates minimal lubricant film around the complete periphery of the discs. Depending on the lubricant and the additives present the extent of the region of low film thickness could vary at the load at which scuffing finally occurred. Ultimately the details of the contact resistance measurements may provide a clearer understanding of the details of the film formation and disappearance process which eventually leads to scuffing.

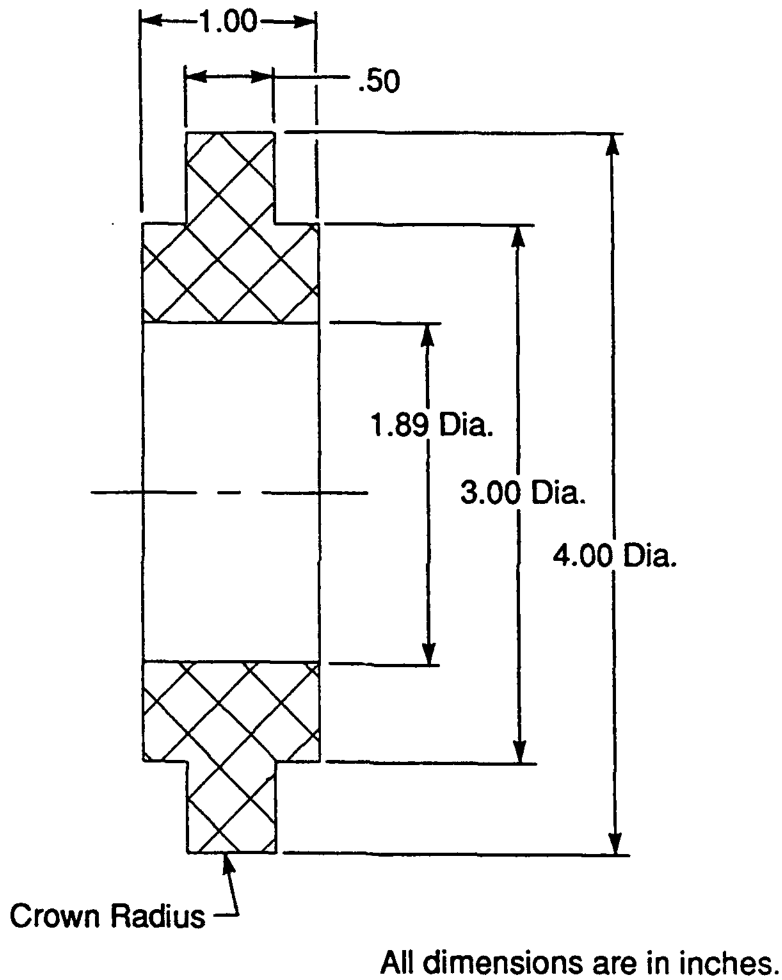
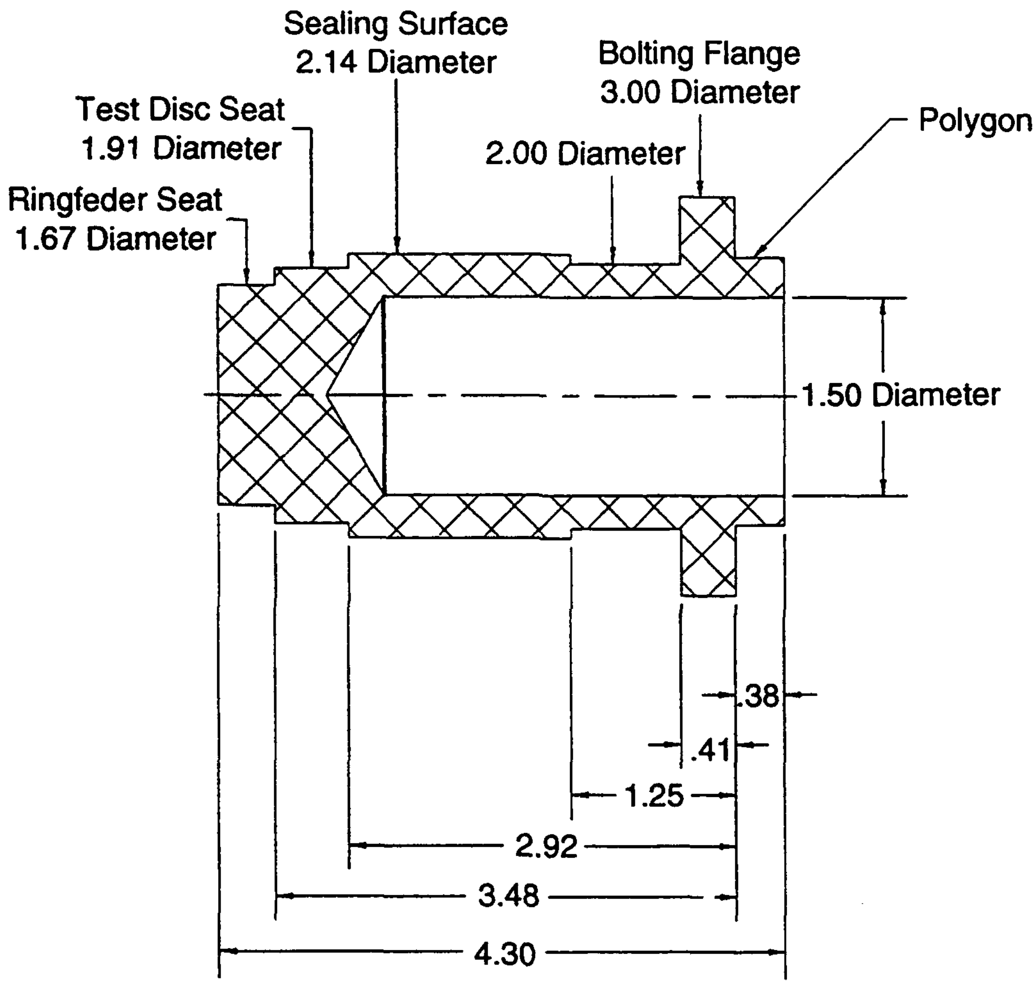


Figure 11 Basic Test Disc Geometry



All Dimensions are in Inches.

Figure 12 Heat Dam Geometry

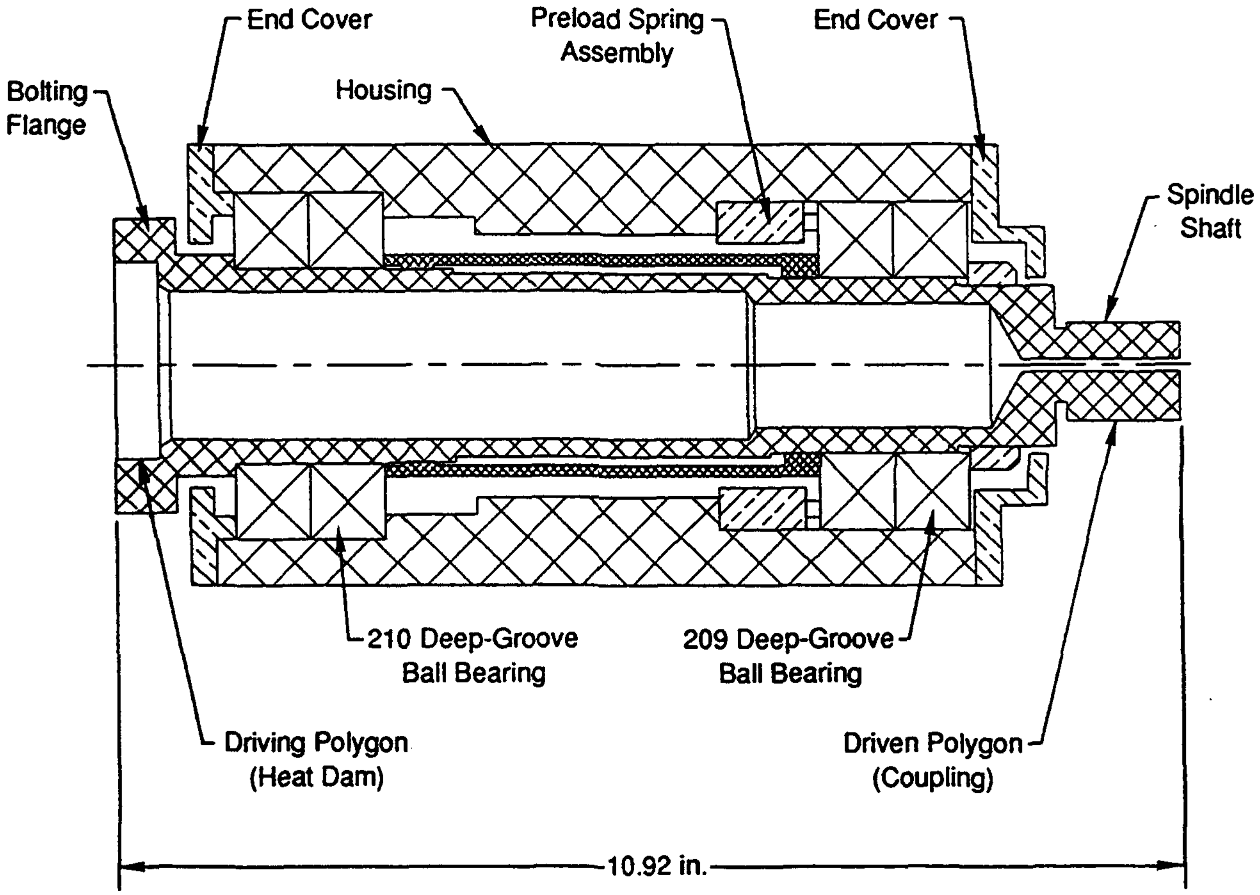


Figure 13 Disc Mounting Spindle Design

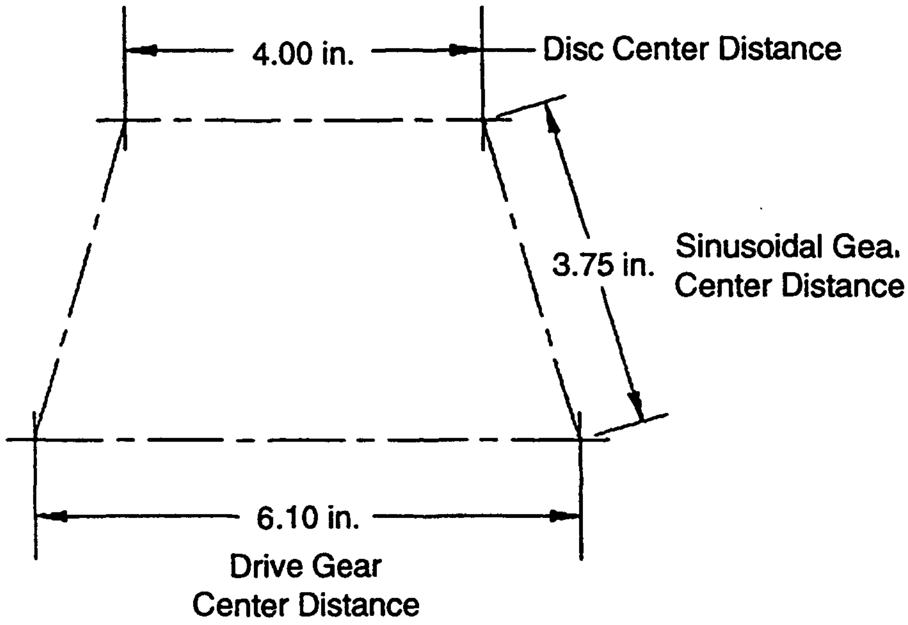


Figure 14 Transmission Shaft Center Lines

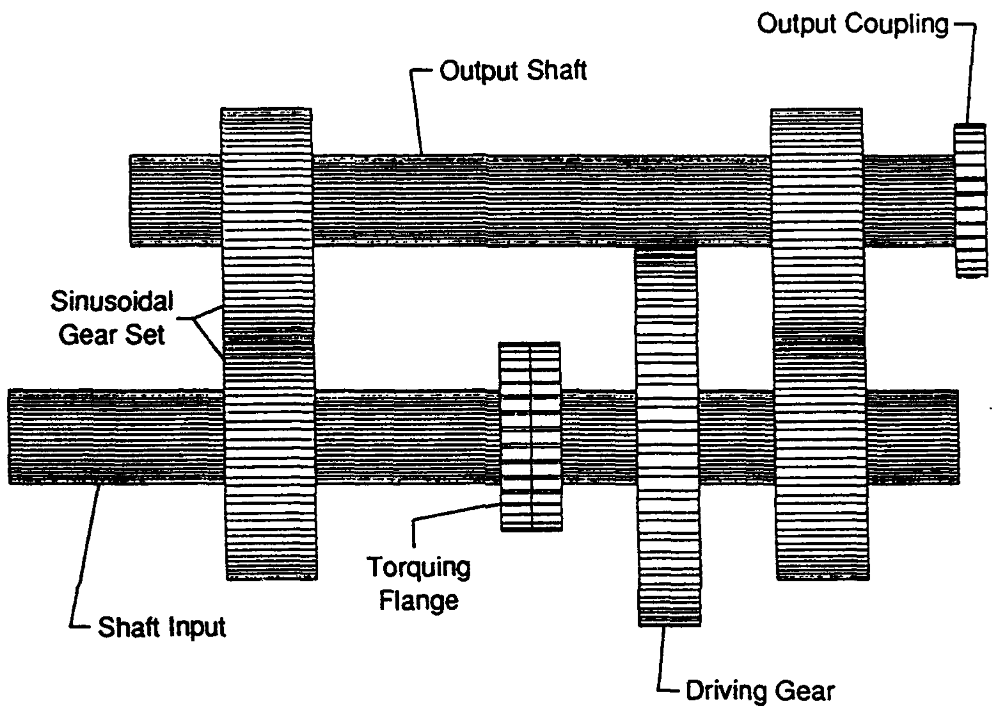


Figure 15 Side View of Transmission Four Square Assembly

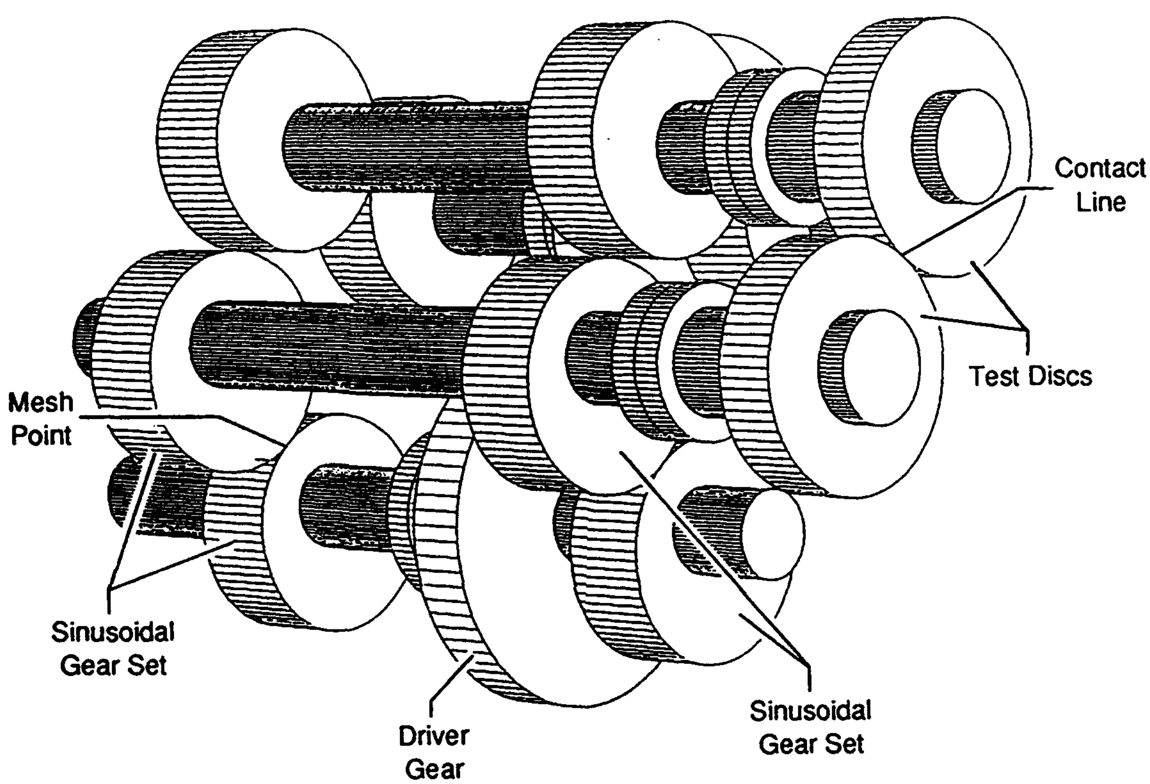


Figure 16 Perspective View of Transmission Gearing

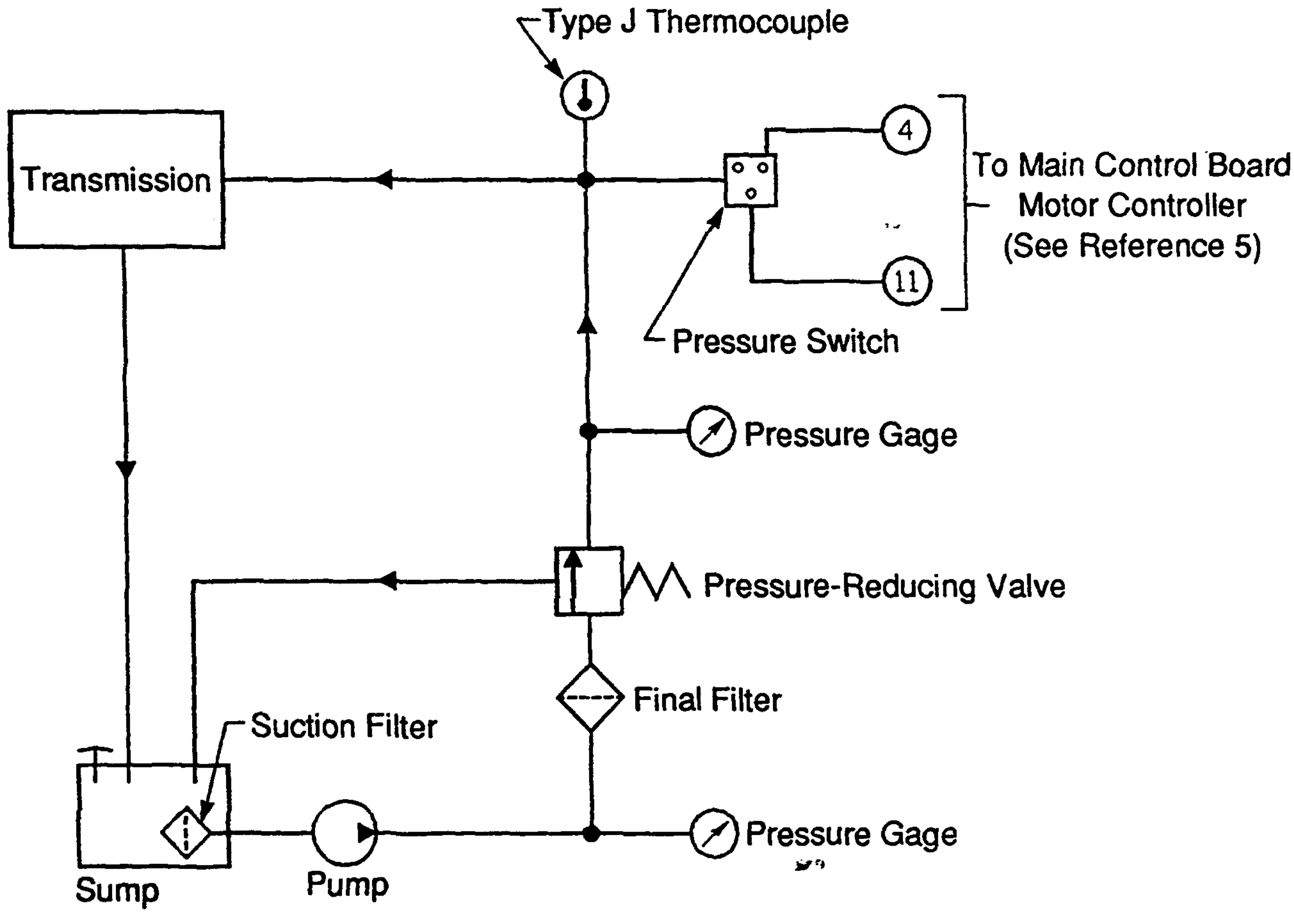


Figure 17 Transmission Lubrication System

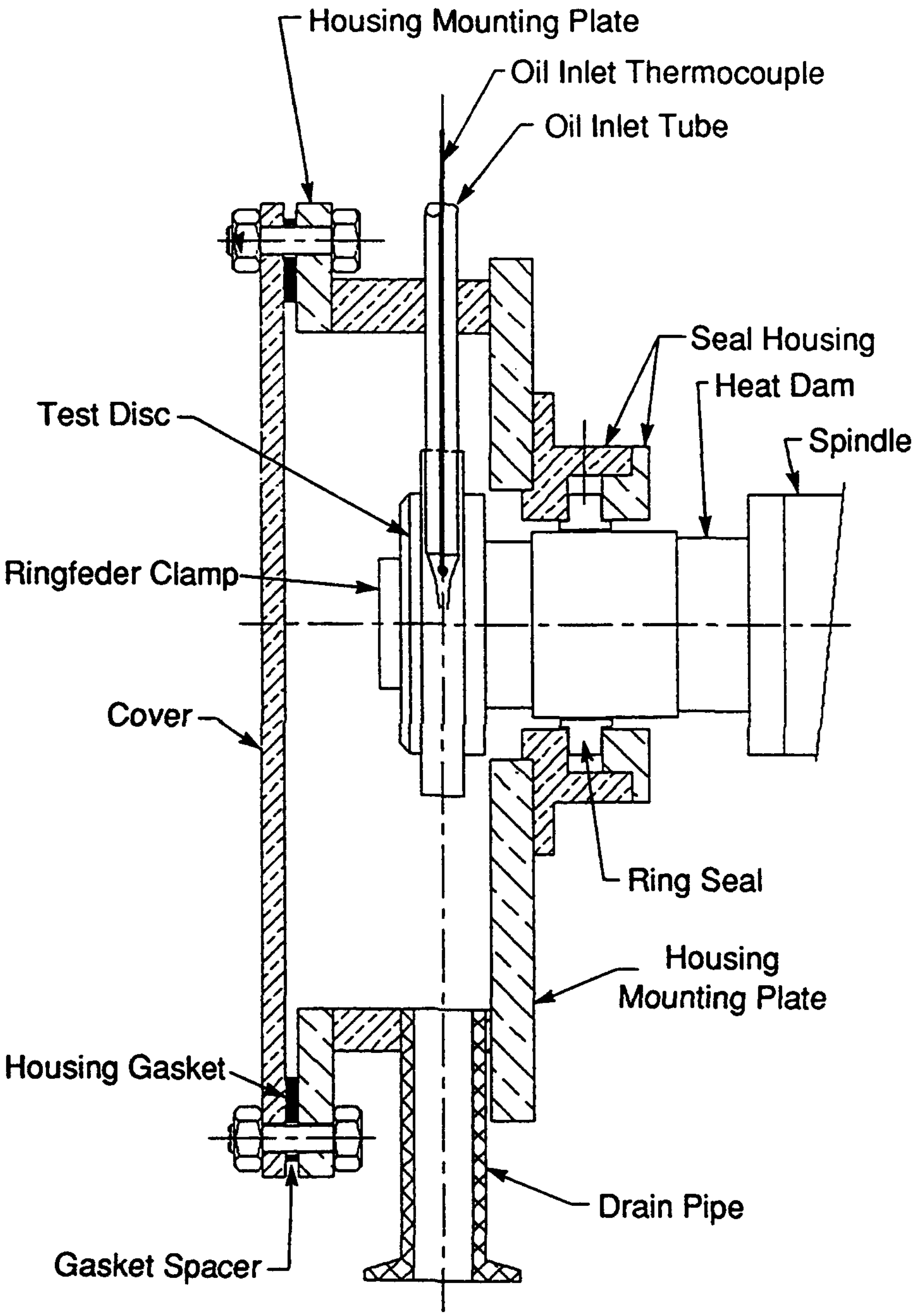


Figure 18 Test Head Assembly

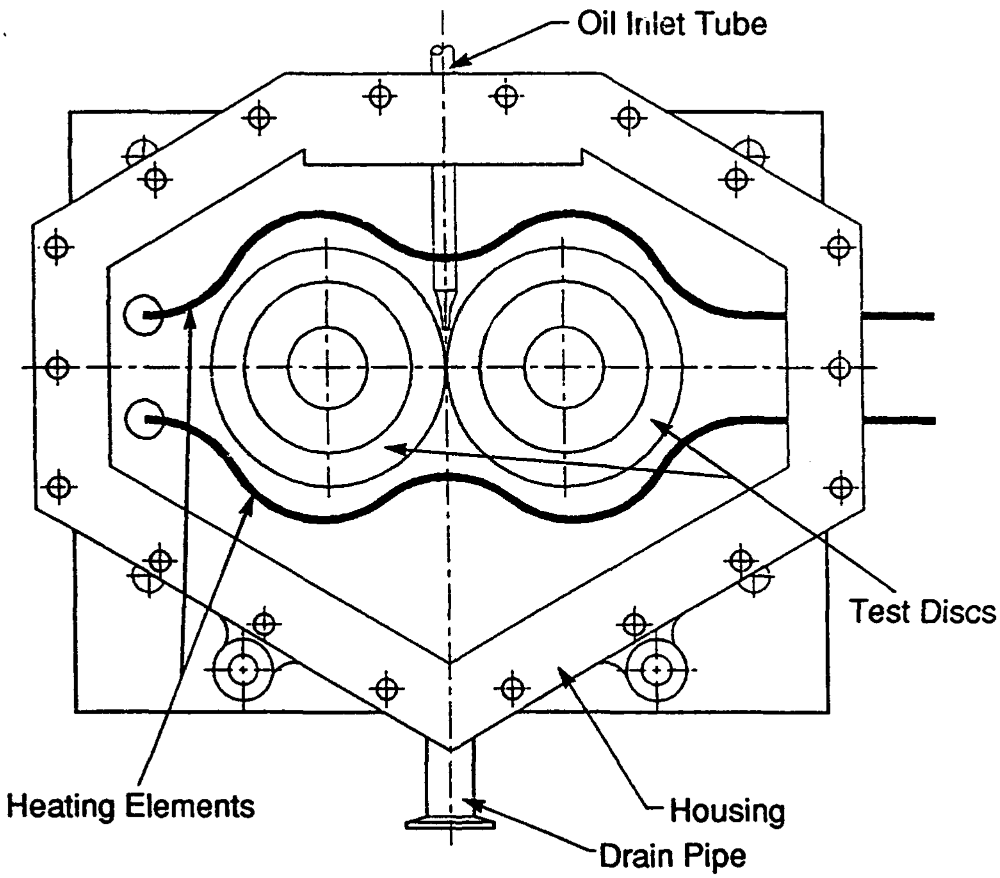


Figure 19 Installation of Test Head Heaters

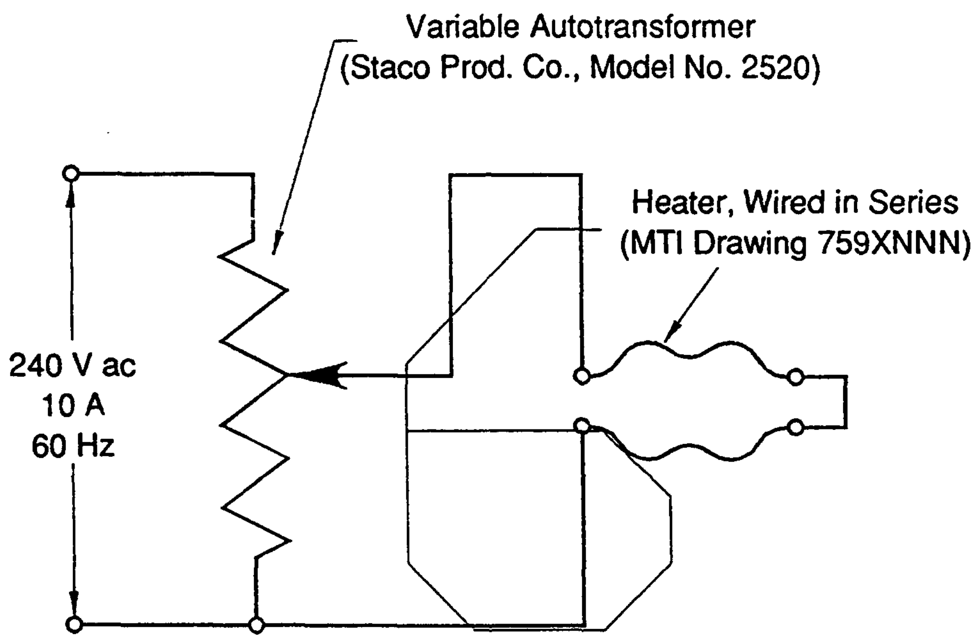


Figure 20 Test Head Heater Electrical Connections

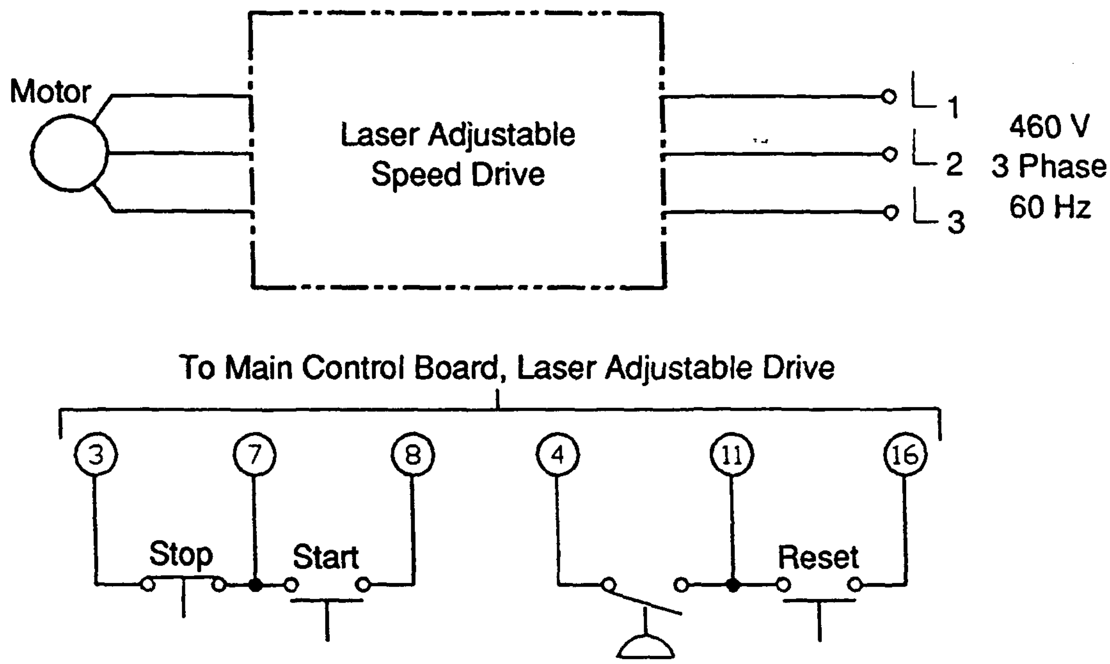


Figure 21 Drive Motor Connection Schematic

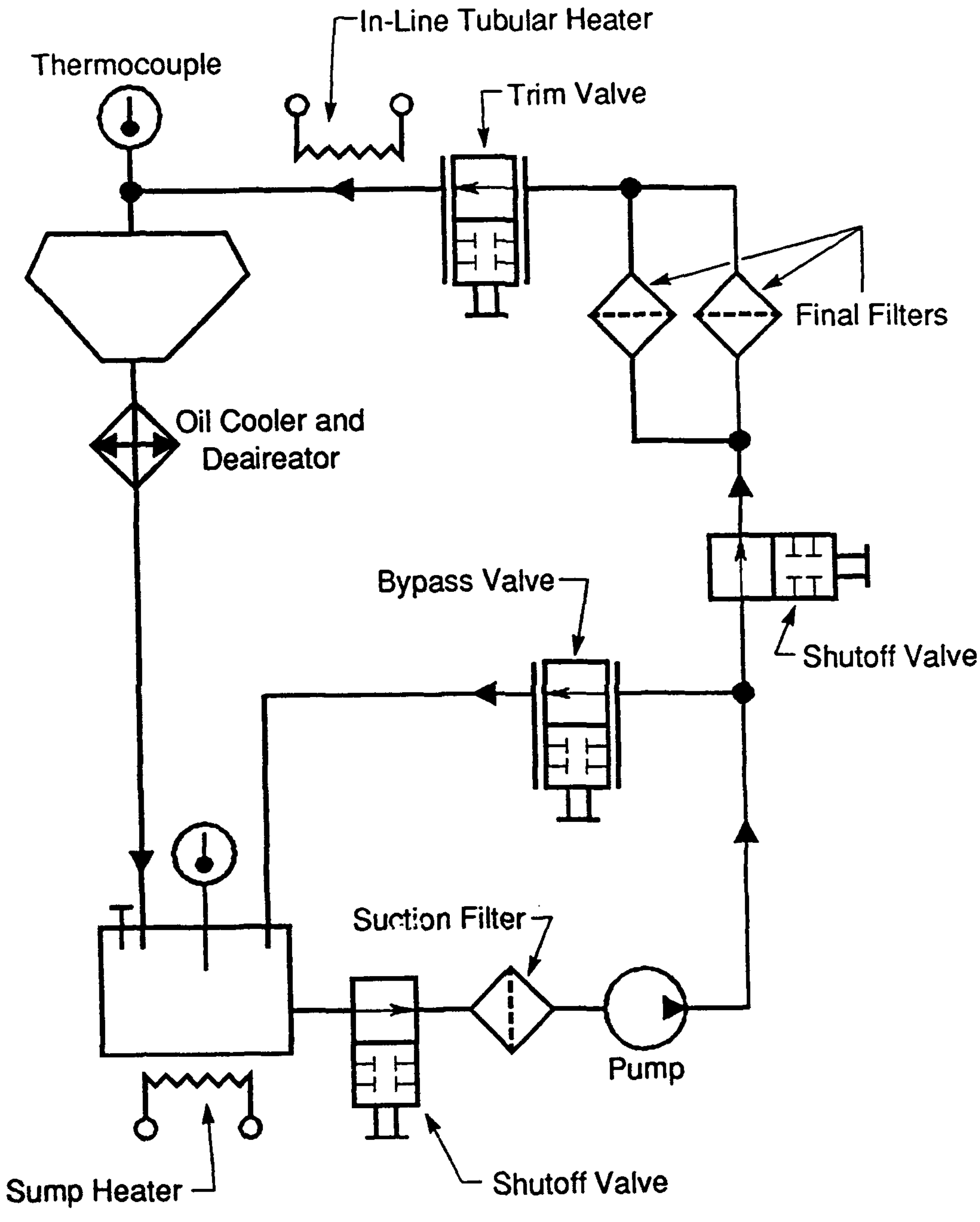


Figure 22 Test Lubricant Supply System Schematic

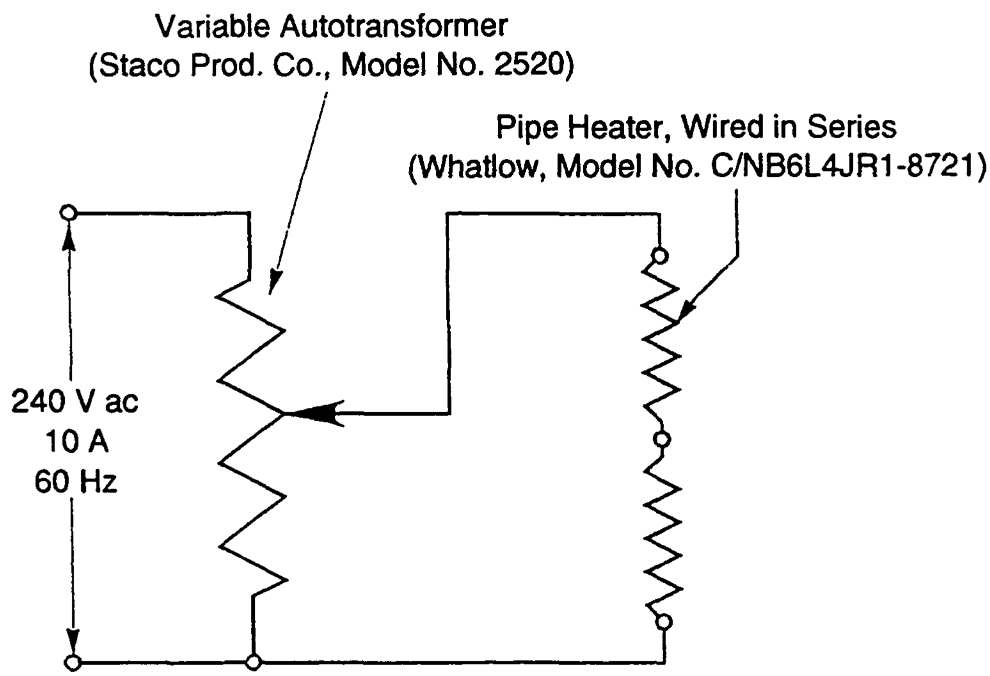


Figure 23 Sump Heater Electrical Connection

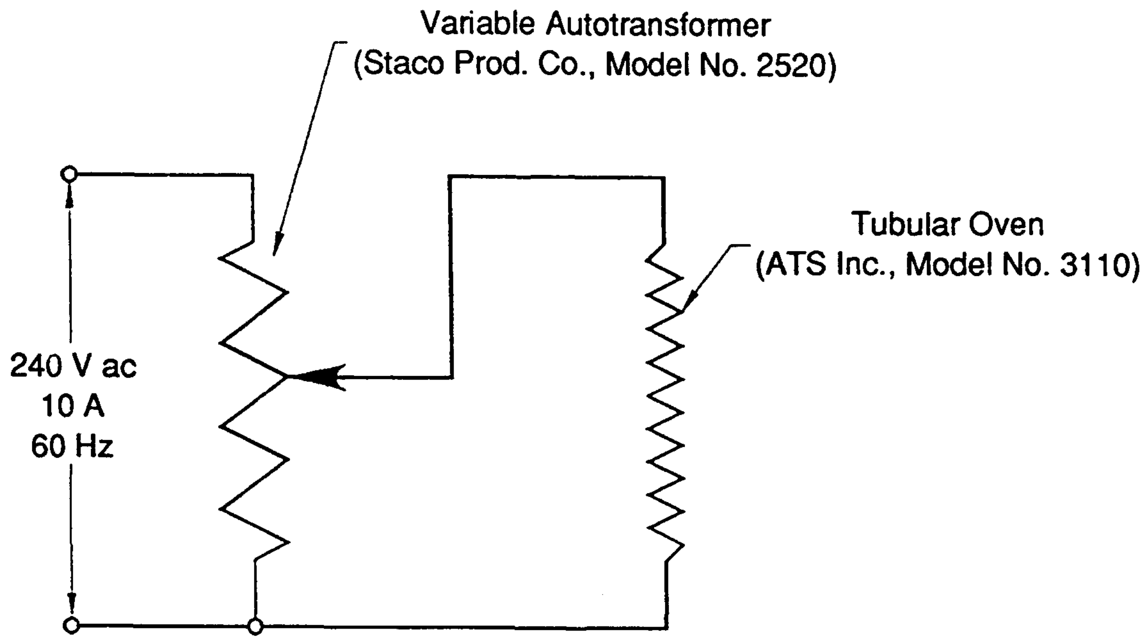


Figure 24 Supply Line Heater Electrical Connection

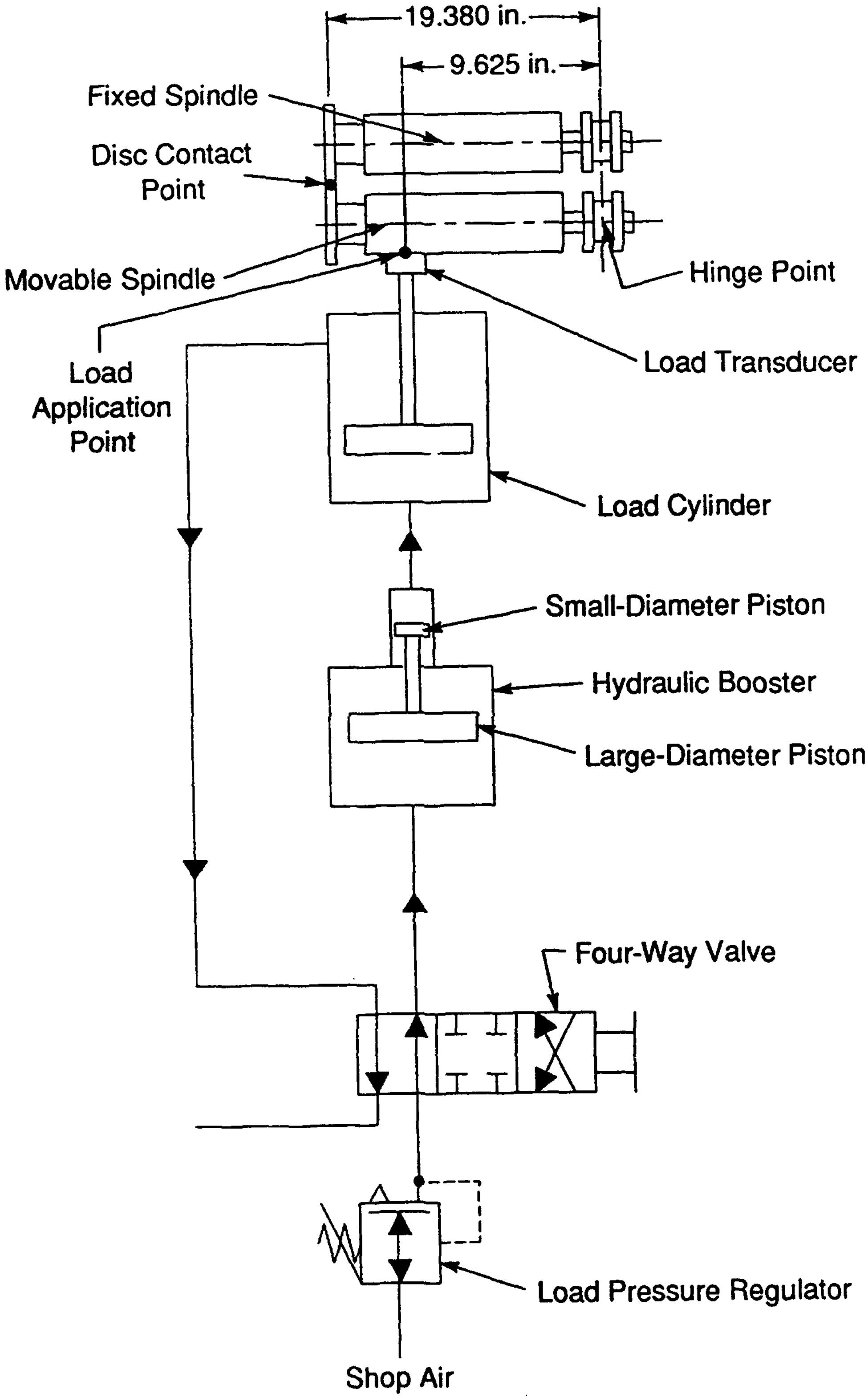


Figure 25 Pneumatic/Hydraulic Loading System Schematic

5.0 TESTING WITH THE TWIN DISC GEAR TOOTH SIMULATOR

Testing for this program was divided into three phases. The first or evaluation phase was designed primarily as a means for debugging the tester and for becoming familiar with its operation. As an adjunct to the familiarization process, the evaluation phase also provided the means for developing a preliminary test procedure to systematize the collection of test data. In the evaluation phase, a single disc geometry was used and most tests were conducted with a single test lubricant.

A second or screening phase was used to examine how various test disc parameters affected the ability of the Twin Disc tester to provide acceptable test data. The preliminary test procedure developed in the evaluation phase was further refined as the screening phase test program was carried out. Screening tests were performed using discs ground with two different crown radii, and at least two different surface finishes. Only one test lubricant was used during these tests.

The final production testing phase was conducted with various test lubricants to produce actual lubricant load capacity test data which could be compared to other load capacity data for the same lubricants from the Ryder gear test. For some test lubricants, data were taken both at the standard Ryder operating temperature (165 F) and at 435 F to evaluate the effects of temperature on load capacity and also the high temperature capabilities of the tester. For standard Ryder temperature data, discs were manufactured from SAE 9310. For the elevated temperature tests, discs were manufactured from Nitralloy N because of its greater temperature capability. This phase used the test procedure developed and defined in the previous test phases with the disc geometry selected from the screening test results. Each of the three test phases are described in the following report sections.

5.1 Evaluation Test Phase

Test results for this phase are shown in Table 1. The first 11 data points were used to prepare a preliminary test procedure, and to learn how to handle both the tester and the test lubricant supply system. The first checks on the test procedure were done at data points E-9 through E-11. The results of these tests were very encouraging, but were taken at a speed where the maximum pressure velocity product (PV) was below the Ryder gear maximum value. The remaining data points were targeted at higher speed testing which produced both the well defined failures at data points E-13 and E-14 and several non-recorded failures resulting from tester malfunctions.

The final test speed selection of 2500 rpm was made because it could produce disc failures at reasonable loads, produce maximum PV products well above the Ryder gear value and left some margin below the maximum design speed of 3300 rpm. The satisfactory conclusion of the evaluation phase led directly to the initiation of the screening phase.

5.2 Screening Test Phase

The intent of the screening test phase was to determine whether there was a specific combination of disc crown radius and surface finish which would provide better test results. Two crown radii and three surface finishes were examined with the test results shown in Table 2. The most consistent data were obtained at data points S2-1 through S2-5 for 14 in. crown discs with surface finishes in the 11 to 14 μin. range. Even the coarser disc finishes showed up well (data points S4-1 through S4-5) with the 14 in. crown. Very smooth discs, when manufactured with good profiles, did not fail. The coarser finished discs and the 18 in. crowned disc with a nominal 12 μin. finish showed more scattered data.

The selection of a 14 in. crown radius and a target surface finish of 12 to 14 μin. for further testing was based in part on the test results obtained from this test sequence and in part on analysis which indicated a 15% reduction in the load required to reach a 200,000 lb/in^2 Hertzian contact stress (felt important in matching Ryder gear values) when using a 14 in. rather than an 18 in. radius.

5.3 Standardized Test Protocol

Based on the results of the screening tests, a standard test protocol was developed. This protocol included procedures for machine setup, instrumentation setup, test lubricant setup, and testing. The use of a standard test procedure was felt to be needed to ensure that as much as possible results would not be affected by differences in procedure. The use of a standard procedure also ensured that all necessary care was taken to avoid mistakes in machine operation which might result in poor data, machine damage or operator injury.

5.3.1 Machine Setup

The machine setup involves a straightforward set of procedures which must be followed to ensure that power is turned on, lubricant is supplied to the gear box, and gas supplies for the seals are turned on. The following is a checklist used by the technician or test engineer during each startup of the Twin Disc machine:

1. Connect power to simulator.
2. Start transmission lubricant pump.
3. Make sure relief valve is set at 40 to 45 psig.
4. Turn on seal gas supply and set pressure at 40 to 50 psig.
5. Make sure the load condition is at 0 lb.
6. Start drive motor.
7. Set speed at 350 to 450 rpm.

5.3.2 Instrumentation Setup

Once the machine has been properly setup for running, the instrumentation is checked, turned on and set for proper conditions. The instrument referred to as the ELECTAC in the write-up is a trade name for the contact resistance measurement system which is used to determine the load at which step loading is initiated.

1. Make sure instrument power is on.
2. Set ELECTAC to the following settings:
   - Contact voltage: 0.1 V
   - Discriminator voltage: 0.05 V
   - Center resistance: 1 kΩ
   - Integrator gain: 1

- Clock: 1 sec

5.3.3 Test Lubricant Start-Up

Once the instrumentation and the machine have been checked and started, the test lubricant system can be started. The lubricant should be circulating while it is heating up to ensure that all portions of the system are heated evenly. The procedures to be followed include the following steps:

1. Make sure the loop shut-off valve (Item 7, Drawing 759E086) is open.
2. Make sure the test lubricant shut-off valve (Item 19, Drawing 759E086) is open.
3. Make sure the pump bypass valve (Item 15, Drawing 759E086) is fully open.
4. Adjust lubricant delivery needle valve (Item 25, Drawing 759E086) fully open.
5. Check sump level. Add lubricant as required. The lubricant supply pump requires a test lubricant viscosity less than 200 cs for safe operation. Any test lubricant not meeting this requirement at room temperature must be heated to a minimum temperature that will provide this viscosity. To accomplish this, the suction line heater between the sump and the pump and the sump heater should be energized to a partial power level commensurate with the required minimum lubricant temperature. When the desired sump temperature is reached, proceed to Step 6.
6. Start test lubricant pump.

Since for tests at the standard Ryder gear test temperature the lubricant is the primary heat source in the machine, it is necessary to run the lubricant system until thermal equilibrium has been achieved in the test head of the machine. This is generally taken as being achieved when the lubricant return temperature to the sump and the supply temperature are constant. A heat up of time of approximately 2 hours is generally used to ensure thermal stability in the tests.

5.3.4 Testing

Testing is conducted by first using the electrical contact resistance to determine the load at which the film first becomes conducting in the high slip region of the disc contact. Once this point has been found, test loads are increased in steps with visual inspections of the disc condition between each step until scuffing is found on the discs. The steps followed in the testing of a lubricant are as detailed below:

1. Adjust load to 0 lb.
2. Set lubricant delivery valve (Item 25, Drawing 759E086) fully open.
3. Adjust lubricant temperature to desired test temperature (±5°F). Adjustment of test lubricant temperature is accomplished with the sump heater (Item 27, Drawing 759E086) and the line heaters (Items 28, 29, and 30, Drawing 759E086). At lubricant temperatures below 200°F, only the sump heater may be required. At lubricant temperatures above 200°F, all heaters may be needed. Do not exceed a sump lubricant temperature reading of 450 to 480°F and line temperature readings of 750°F.
4. Adjust speed to 2500 rpm.

5. Adjust load to obtain a minimum ASPERITAC (ELECTAC) reading of approximately 75 to 85%. Adjust load no higher than the level at which a 100% reading is first obtained.
6. Run simulator for 2 min. and record load and reading on data sheet*.
7. Adjust load upward to obtain an ASPERITAC (ELECTAC) reading of approximately 85 to 95%. Run load for 2 min. and record data. *Under certain conditions, Step 7 will immediately produce a reading of 100%. If this occurs, go to Step 9. Early failure detection will occur when the contact count readings start to increase at a fixed or increasing load.*
8. Repeat Step 7 until load reading is consistently at or above 95%.
9. When reading remains above 95 to 100% for more than 2 min., remove the load, stop the simulator, and examine the discs.
10. Proceed as follows if discs are:
    a. Scuffed - terminate testing.
    b. Not scuffed - resume speed and increment load by 30 to 60 lb. Run for 2 min. Remove the load, stop the simulator, and examine the discs. If discs are not scuffed repeat Step 10b: if scuffed, terminate test.

In all of the production testing discussed below the test protocol described in this section was used. For the elevated temperature testing longer times were given during the heating up of the rig to ensure that thermal equilibrium had been reached.

5.4 Production Test Phase

The last experimental phase of the Twin Disc tester development program was the production test phase. In this phase, the test procedure and disc geometry parameters selected from the earlier test work were applied to the testing of a range of lubricants supplied by the Air Force. The make-up of these lubricants was unknown (only code designations were indicated) at the time they were tested.

The purpose for the production phase testing was to ascertain whether the Twin Disc tester would rank the test lubricants relative to load capacity as they were ranked by the Ryder gear tester. The results of the production testing on the Air Force supplied lubricants is shown in Table 3. In total, at least 3 tests were run on 10 different lubricants with 5 lubricants run at two temperatures, for a total of 49 tests. A discussion of the production test results and recommendations for the Twin Disc tester will be found in the following report sections.

5.5 Navy Lubricant Production Testing

In order to obtain further information on the relative ranking of lubricants and the general usefulness of the Twin Disc machine, a series of tests were run on lubricants provided by NAWC Trenton. These studies evaluated a combination of standard Navy MIL-L-23699 type lubricants,

* A data sheet that you may copy for this purpose is provided at the end of this section.

reference lubricants, and high load capacity gear lubricants. The results of these tests are given in Table 4.

During the running of these tests, an error was made in the disc material used with two of the lubricants. In the case of lubricants 13 and 15 several of the tests were run using Nitralloy N discs instead of the standard 9310 material. Because the load capacity numbers can be affected by the disc material used, these lubricants were reevaluated with reground 9310 discs.

For lubricant 15, the rerun results were similar to those obtained in the original tests of this lubricant using both 9310 and Nitralloy discs. For lubricant 13, however, much lower and scattered values were obtained during the rerunning of these tests. There is still a question about the values for this lubricant in this test. While reground discs have been used successfully previously with other lubricants, and materials evaluation of scuffed discs has shown that the depth of damage is small after a scuff occurs, the extremely low values in some cases in Table 5 leave these results in question. Because of the unavailability of additional test discs it was not possible to rerun these tests a third time in the hopes of obtaining more consistent results.

Table 1 Evaluation Testing Conditions and Results

Data Point	Speed (rpm)	Lubricant Identifier	Left Disc	Right Disc	Failure Load(lb.)	Comments on Test
E-1	1627	8105				Did not attempt to achieve failure
E-2	1627	8105				Did not attempt to achieve failure
E-3	1627	8105				Did not attempt to achieve failure
E-4	1627	9041				Did not attempt to achieve failure
E-5	1627	9041				Did not attempt to achieve failure
E-6	1627	9041				Did not attempt to achieve failure
E-7	1627	9041				Did not attempt to achieve failure
E-8	1627	9041				Did not attempt to achieve failure
E-9	1627	9041	E4	E3	1290	
E-10	1627	9041	E1	E2	1210	
E-11	1627	9041	E6	E9	1280	
E-12	2160	9041	E6	E9		Did not attempt to achieve failure
E-13	2500	9041	E13	E14	1106	
E-14	2500	9041	E5	E10	1163	
E-15	2500	9041	E7	E8		Slip ring malfunction
E-16	350	9041	E11	E12		Break-in only
E-17	2500	9041	E11	E12		Lost test lubricant pump power

Table 2 Results of Screening Tests

Data Point	Left Disc	Right Disc	Crown R (in.)	Finish ($x10^{-6}$ in.)	Failure Load (lb.)	Comments on Test
S1-1	P1	P2	18	5		No failure at max. load
S1-2	P7	P8	18	4		No failure at max. load
S1-3	P1	P2	18	5		No failure at max. load
S2-1	E3	E5	14	14	1039	
S2-2	E4	E8	14	13	1002	
S2-3	E9	E14	14	12	1003	
S2-4	E13	E7	14	11	980	
S2-5	E6	E10	14	12	1450	
S3-1	P9	P2	18	12	975	
S3-2	P3	P4	18	12	1259	
S3-3	P6	P5	18	11		No failure at max. load
S3-4	P10	P7	18	12	1221	
S3-5	P8	P1	18	12	525	
S4-1	P3	P4	14	23	1200	
S4-2	P6	E8	14	23	1260	
S4-3	E14	E3	14	23	930	
S4-4	E6	E10	14	24	954	
S4-5	P5	E9	14	23		No failure at max. load

Table 3a Production Testing Results

Data Point	Lubricant Identifier	Test Temp (F)	Left Disc	Right Disc	Finish ($x10^{-6}$ in.)	Failure Load (lb.)
P1-1	90086	165	N3	N1	19	953
P1-2	90086	165	N2	N5	20	1086
P1-3	90086	165	N4	N6	21	1068
P2-1	90080	165	P14	P13	20	795
P2-2	90080	165	P11	P12	20	550
P2-3	90080	165	P10	P5	24	867
P3-1	90079	165	P2	P7	23	329
P3-2	90079	165	P1	P8	24	233
P3-3	90079	165	P4	P9	25	113
P4-1	90082	165	P3	P6	20	720
P4-2	90082	165	P15	P16	20	712
P4-3	90082	165	P17	P18	20	766
P5-1	90086	435	NR1	NR2	23	396
P5-1	90086	435	NR3	NR4	25	377
P5-3	90086	435	NR5	NR7	25	363
P6-1	90084	165	R1	R2	12	992
P6-2	90084	165	R3	R4	12	735
P6-3	90084	165	R5	R6	13	678
P7-1	8105	165	R7	R8	13	1066
P7-2	8105	165	R9	R10	16	900
P7-3	8105	165	R11	R12	17	797
P8-1	90079	165	R13	R14	15	696
P8-2	90079	165	R15	R16	16	633
P8-3	90079	165	R17	R18	18	631

Table 3b Production Testing Results

Data Point	Lubricant Identifier	Test Temp (F)	Left Disc	Right Disc	Finish ($x10^{-6}$ in.)	Failure Load (lb.)
T1-1	90032	166	2P11	2P12	21	540
T1-2	90032	164	2P13	2P14	22	537
T1-3	90032	168	2R1	2R2	19	781
T1-4	90032	166	2R3	2R4	20	650
T2-1	90032	432	2N1	2N2	19	126
T2-2	90032	435	2N3	2N4	15	126
T2-3	90032	437	2N5	2N6	17	124
T3-1	90033	169	2R5	2R6	20	655
T3-2	90033	166	2R7	2R8	21	749
T3-3	90033	168	2R9	2R10	22	600
T7-1	90033	429	2A-7	2A-8	27	227
T7-2	90033	431	2A-9	2A-10	17	276
T7-3	90033	429	2A-11	2A-12	19	221
T4-1	90034	167	2R11	2R12	25	913
T4-2	90034	168	2R13	2R14	21	913
T4-3	90034	169	2R15	2R16	22	811
T8-1	90034	448	2A-13	2A-14	19	173
T8-2	90034	446	2A-15	2A-16	22	326
T8-3	90034	427	2A-17	2A-18	16	275
T5-1	90035	169	2R17	2R18	22	NONE AT 1560
T5-2	90035	168	2R19	2R20	18	NONE AT 1560
T5-3	90035	165	2E3	2E4	21	NONE AT 1560
T6-1	90035	433	2A-1	2A-2	25	578
T6-2	90035	428	2A-3	2A-4	17	273
T6-3	90035	433	2A-5	2A-6	16	378

Table 4 Navy Lubricant Evaluation Results

Data Point	Lubricant Identifier	Test Temp (F)	Left Disc	Right Disc	Finish ($x10^{-6}$ in.)	Failure Load (lb)
N-1	11	164	NA-1	NA-2	15	1109
N-2	11	165	NA-3	NA-4	20	907
N-3	11	166	NA-5	NA-6	13	695
N-4	11	166	NA-7	NA-8	18	755
N-5	11	166	NA-9	NA-10	22	700
N-6	12	166	NA-11	NA-12	18	NONE AT 1560
N-7	12	167	NA-13	NA-14	19	NONE AT 1560
N-8	12	166	NA-15	NA-16	17	NONE AT 1560
N-9	12	166	NA-17	NA-18	20	NONE AT 1560
N-10	12	168	NA-19	NA-20	18	1282
N-11	13	164	NA-21	NA-22	15	801
N-12	13	165	NA-23	NA-24	16	798
*N-13	13	165	2A-1 *Nit	2A-2	13	543
*N-14	13	165	2A-3 *Nit	2A-4	16	537
*N-15	13	164	2A-7 *Nit	2A-8	14	587
*N-16	14	166	2A-9 *Nit	2A-10	16	798
N-17	14	166	E3-2R	E4-2R	13	797
N-18	14	166	P6-2R	E2-2R	20	906
N-19	14	166	E9-2R	E13-2R	21	850
N-20	14	165	E7-2R	E14-2R	20	956
N-21	15	165	P11-2R	E5-2R	18	744
*N-22	15	169	N1-R *Nit	N4-R	15	383
*N-23	15	164	2A-11 *Nit	2A-12	13	586
*N-24	15	165	2A-19 *Nit	2A-20	15	585
N-25	15	165	P13-2R	R17-2R	16	593

* and *Nit indicate that these results erroneously used Nitralloy discs instead of standard 9310

Table 5 Navy Lubricant Evaluation Results of Retest on Lubricants 13 and 15

Data Point	Lubricant Identifier	Test Temp (F)	Left Disc	Right Disc	Finish ($x10^{-6}$ in.)	Failure Load (lb.)
N-11R	13	166	RW-1	RW-2	14.6	486
N-12R	13	168	RW-3	RW-4	12.4	431
N-13R	13	163	RW-5	RW-6	12.8	171
N-14R	13	163	RW-7	RW-8	16.3	125
N-15R	13	170	RW-9	RW-10	12.8	123
ADDED PT.	13	165	RW-18	RW-19	12.9	481
ADDED PT.	13	165	RW-17	RW-20	14.9	224
N-22R	15	166	RW-11	RW-12	13.5	534
N-23R	15	164	RW-13	RW-14	15.3	533
N-24R	15	165	RW-15	RW-16	15.5	535

6.0 DISCUSSION OF RESULTS

In reviewing the results of the scuffing tests conducted to date, an obvious comparison is to evaluate the Twin Disc results with respect to the Ryder gear results for the same lubricants. This section discusses that comparison where data have been made available by the Air Force and the Navy. The second portion of this section summarizes the comparisons between the test conditions in the Twin Disc and the Ryder gear and discusses some hypotheses about why lubricants may be ranked differently by the two machines in some cases.

In all of the tests it was necessary to use visual inspection to verify scuffing of the discs. Early efforts to use electrical contact resistance were not successful since they showed that even at very low voltages the contact resistance becomes zero (indicating significant asperity contact) at loads significantly below the ultimate scuffing load. This difference between the existence of load conditions where there is essentially a minimal lubricant film as measured by contact resistance, and the load at which scuffing occurs was particularly large with fully formulated lubricants where boundary lubrication is expected to be much better than in a pure basestock lubricant.

6.1 Comparison with Ryder Gear Data

The scuffing test results listed chronologically in Table 3 can be ranked according to the average failure load for each test lubricant. The resulting rankings are shown in, Table 6 for 165°F tests and Table 7 for 435°F tests.

Two tests, Sequences P1 and P5, compare the same material at two lubricant inlet temperatures (165°F and 435°F) and show a load capacity loss of 64% due to temperature.

Test sequences T1 and T2, T3 and T7, T4 and T8, T5 and T6 compare the appropriate disc material for each of two inlet lubricant temperatures (9310 @ 165°F and NIT-N @ 435°F) and show losses of 35%, 64% and 71%. The comparison for T5 and T6 cannot be realistically made because the T5 tests showed no failure for either the Twin Disc or Ryder gear tester at their respective maximum load capability (1560 lb. for the Twin Disc machine, 5260 lb. for the Ryder gear tester).

The final comparison to be made is between the rankings of the test lubricants based on the Twin Disc tests to the rankings of the same lubricants taken on the traditional Ryder gear tester. This comparison is shown in Table 8.

Although the tabularized ranking comparisons appear scattered, the variation in load between many ranks is very low. In a graphical comparison shown in Fig. 26, the scattering of rankings does not seem quite as diffused as the tables make the data appear. A 100 pound variation in the failure load reported in either test could make a significant difference in the comparison. The major differences between test methods appears to be the twin disc failure loads at P1, P2 and P8.

For tests in sequence P1, the high readings were very consistent, varying by no more than 8%. The possible explanation for the high Twin Disc loads at these test points are either that this particular lubricant is more resistant to scuffing during the slow transition from pure rolling to pure sliding of the Twin Disc machine than it is for the more rapid transition experienced on the Ryder gear tester (less heat loss and possibly a higher than expected Blok critical temperature or differences in overall bulk temperatures, or there is something related to the disc finish that promotes better load carrying capacity. A more detailed look at these disc surface finishes was not possible because at the time of the P1 tests, detailed finish and contour measuring instruments were not available.

The lower than expected Twin Disc failure loads at test sequences P2 and P8 may very well result from the same causes, but because of the particular lubricant formulation and the additives used, the observed differences were opposite to the P1 test results.

The P8 test data, taken with essentially correct discs, were very consistent, showing no more than a 6½% variation in failure loads, but for reasons not yet determined, the failure loads were substantially lower than Ryder gear test data. Earlier tests using the same lubricant (P3 test series) had even lower failure loads. The P3 tests were run with very coarse discs which might account for the lower loads. The P2 tests were also run with very coarse discs and the lubricant they were tested with may not have been able to handle the effect of larger asperities.

Even with the early use of the contact resistance measuring instrument, the criteria for failure was to examine the discs after a period of running at each load increment. After this instrumentation method proved unusable, the visual observation of scuffing failure continued.

Surface finish of each test crown disc was always measured prior to testing with the desired finish for production test phase in the range of 11 to 14 μin. RA. Although this was the target finish, many test discs had much coarser finishes. This coarser finish did not seem to present a testing problem based on the data from screening test series S5 which indicated both high loads and good repeatability.

On this basis, it was decided to test the coarser finishes rather than spend the time and effort in any rework. What was not considered at the time, but may have contributed to the poor showing of some tests, particularly for the P2 and P8 tests, both of which were run with axial finish levels at the 25 μin. level is the actual crown contour shape. An instrument to measure this characteristic became available to the program during the P2 test series. Two examples of different crown shapes can be seen in Figures 27 a and b. While the two discs were measured as having comparable surface finishes, but the crown contour on disc 2A-1 is much different than the crown on disc P9. On disc P9 grooving in the grinding wheel has been transferred to the disc resulting in a long wavelength surface irregularity which isn't measured in a surface finish measurement. Clearly both disc finish and contour must be measured to insure proper control of the test results. The more irregular contour of the P9 points to a possible factor in the lower than expected scuffing loads experienced at data points such as P2, P3 and P8 since the irregularities can lead to locally higher PV products than the overall PV product assumes. Clearly improved disc manufacturing control is needed if Twin Disc is to be used for routine lubricant evaluation.

After completion of the tests of Air Force lubricants, a series of tests were conducted on five lubricants supplied by the Navy (NAWC, Trenton). The results of these Twin Disc tests were much more consistent with the Ryder tests for the same lubricants as shown in Table 9. If the lubricant which exceeded machine capabilities in both the Twin Disc and Ryder tester is assigned the highest ranking, then all five lubricants are ranked in the same order by the two machines. This correlation seems to verify that while differences in ranking can be found for lubricants with similar performance, both machines can provide usable relative performance measures for lubricants and clearly distinguish between poor performing lubricants, adequate lubricants and high load capacity materials. Based on the data taken to date, there is also some indications that the Twin Disc rig may spread the failure data for similar lubricants over a wider load range providing more sensitivity in evaluating formulation effects in lubricants.

6.2 Review of Comparison of Twin Disc and Ryder Operating Conditions

In reviewing the data, it can be useful to recall the comparisons and trade-offs which have been in designing the Twin Disc rig to simulate the conditions in the Ryder gear tester. Table 10 shows the conditions which are present both at the maximum safe operating conditions in the Twin Disc and at the standard operating conditions used in the majority of the tests reported here. A review of this comparison shows reasonable relative comparison between the basic loads, speeds (both rolling and sliding), and stresses imposed on the lubricant by the two machines. One area, however, where there is a significant difference is in the time scale of the occurrence of the region of high sliding and consequently the high flash or Blok temperature which is generally thought to cause scuffing. The difference in the time scales for the gear and disc is over an order of magnitude if the time to go from pure rolling to max flash temperature is compared. For the disc rig, the time for significant flash temperature effects is probably less than the time to go from rolling to max sliding since flash temperature effects below some value probably have minimal effect on the lubricant.

Figure 28 is a plot comparing the flash temperature cycles of the Twin Disc and the Ryder gear. For the twin disc, one complete cycle of the discs is plotted showing the sinusoidal variation of flash temperature with time. For the Ryder gear the flash temperature cycle for one tooth is plotted. For each tooth two peaks occur in flash temperature as the gear teeth slide as they enter contact and then slide again as they exit contact. At the standard Ryder speed of 10,000 rpm, the same tooth will see its second cycle in roughly the same time interval as it takes the Twin Disc to reach its first temperature maximum. In total, the same tooth will be going through its fifth contact cycle at the time when the twin disc has completed one full cycle of the discs. For the sake of demonstration the maximum temperatures have been set the same in each case.

Clearly the heat transmission and heat build up in the disc and gear are also significantly different. As has been shown by Olver [6], the scuffing performance of a lubricant depends not only on the Blok flash temperature but also on the bulk operating temperature of the disc or gear which tends to become significantly different from the lubricant supply temperature as the loads at which scuffing occur increase and consequently the contact heating in the disc or gear also increases. While a detailed thermal analysis has not been done to compare the relative bulk and flash (Blok) temperatures of the two tests, Fig. 29 indicates conceptually how the two could vary resulting in different ranking of different lubricants.

In the complex interaction in the two machines, it could be likely that the discs in the Twin Disc operate at a higher bulk temperature than the Ryder gears because they are sliding over a longer period of time. This might explain the very low results for basestock lubricants. At the same time because of the higher bulk temperature and the larger region (and hence longer time) over which the flash temperature is high in the Twin Disc, fully formulated lubricants might perform better relative to basestocks because temperature sensitive additives would have more time to react and protect the disc surfaces than they would at the much shorter interaction times in the Ryder gear rig. If this can be shown to be true, it would indicate that the Twin Disc rig might be a more sensitive test for evaluating additive effectiveness than the Ryder gear test. Much more detailed measurements of disc and gear temperatures (possibly using IR techniques) and thermal analyses or the two machine would be needed to truly understand the differences which have been observed.

Table 6 Ranking of Test Lubricants @ 165°F

Rank	Test Lubricant	Disc Matl.	Avg. Failure Load (lb.)	Test Seq.
1	90035	9310	Above 1560 lbs.	T5
2	90086	NIT-N	1036	P1
3	8105	9310	921	P7
4	90034	9310	879	T4
5	90084	9310	802	P6
6	90080	9310	737	P2
7	90082	9310	733	P4
8	90033	9310	668	T3
9	90079	9310	653	P8
10	90032	9310	627	T1

Table 7 Ranking of Test Lubricants @ 435°F

Rank	Test Lubricant	Disc Matl.	Avg. Failure Load (lbs.)	Test Seq.
1	90035	NIT-N	410	T6
2	90086	NIT-N	378	P5
3	90034	NIT-N	258	T8
4	90033	NIT-N	241	T7
5	90032	NIT-N	125	T2

Table 8 Twin Disc to Ryder Scuffing Failure Comparison

Lubricant Designation	Ryder Rank	Twin Disc Rank
90034	1	1
90080	2	6
90079	3	9
8105	4	3
90034	5	4
90033	6	8
90082	7	7
90084	8	5
90086	9	2
90032	10	10
90086*	11	11

*Hot Test, Ryder Test @ 425°F, Twin Disc Test at 435°F

Table 9 Twin Disc to Ryder Scuffing Failure Comparison for Navy Lubricants

Lubricant Designation	Ryder Rank	Twin Disc Rank
12	R	R
14	1	1
11	2	2
15	3	3
13	4	4

Table 10 Comparison of Twin Disc and Ryder Gear Operating Conditions

PARAMETER	TWIN DISC DESIGN	RYDER GEAR VALUE
Average Rolling Speed	701 in/sec(at 3347 rpm)	701 in/sec (at 10,000 rpm)
Max sliding speed	304 in/sec	585 in/sec
Sliding speed at Max PV	304 in/sec	363 in /sec
Max Hertz pressure	202,085 psi (1000 lb/18 in)	215,000 psi
Max load	1600 lb.	755 lb
Max PV (lb x in/sec)	304,000 (1000 lb)	162,000
Max EHD film (microin.)	18.8	13.4
Blok Temperature	283 F (1000 lb/3347 rpm)	148 F
Contact Flash Time	$4.5x10^{-3}$(0-max)/$9x10^{-3}$(0-0)	$1x10^{-4}$ (0-0)
Contact Repeat period	$18x10^{-3}$	$6x10^{-3}$
Surface finish (microin.)	12-14	18-25

PARAMETER	TWIN DISC RUNNING	RYDER GEAR VALUE
Average Rolling Speed	523 in/sec(at 2500 rpm)	701 in/sec (at 10,000 rpm)
Max sliding speed	227 in/sec	585 in/sec
Sliding speed at Max PV	227 in/sec	363 in /sec
Max Hertz pressure	214,606 psi (1000 lb/14 in)	215,000 psi
Max load	1600 lb.	755 lb
Max PV (lb x in/sec)	227,000 (1000 lb)	162,000
Max EHD film (microin.)	15.0	13.4
Blok Temperature	211 F (1000 lb/2500 rpm)	148 F
Contact Flash Time	$6x10^{-3}$(0-max)/$12x10^{-3}$(0-0)	$1x10^{-4}$
Contact Repeat period	$24 x10^{-3}$	$6x10^{-3}$
Surface finish (microin.)	12-14	18-25

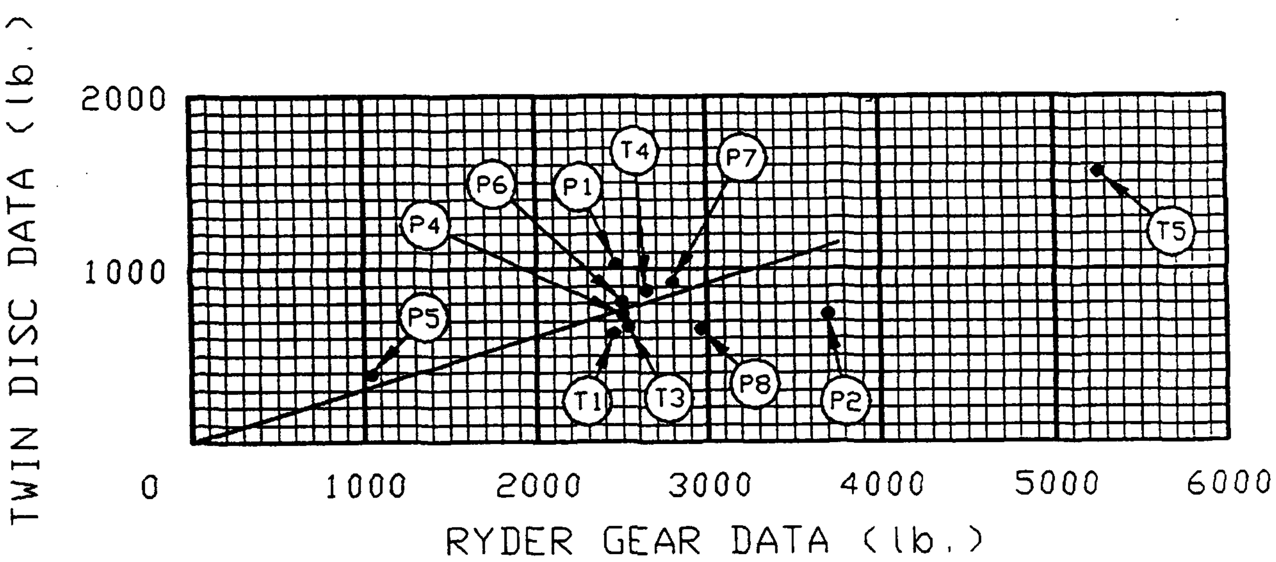


Figure 26 Comparison of Ryder and Twin Disc Scuffing Values for Air Force Lubricants

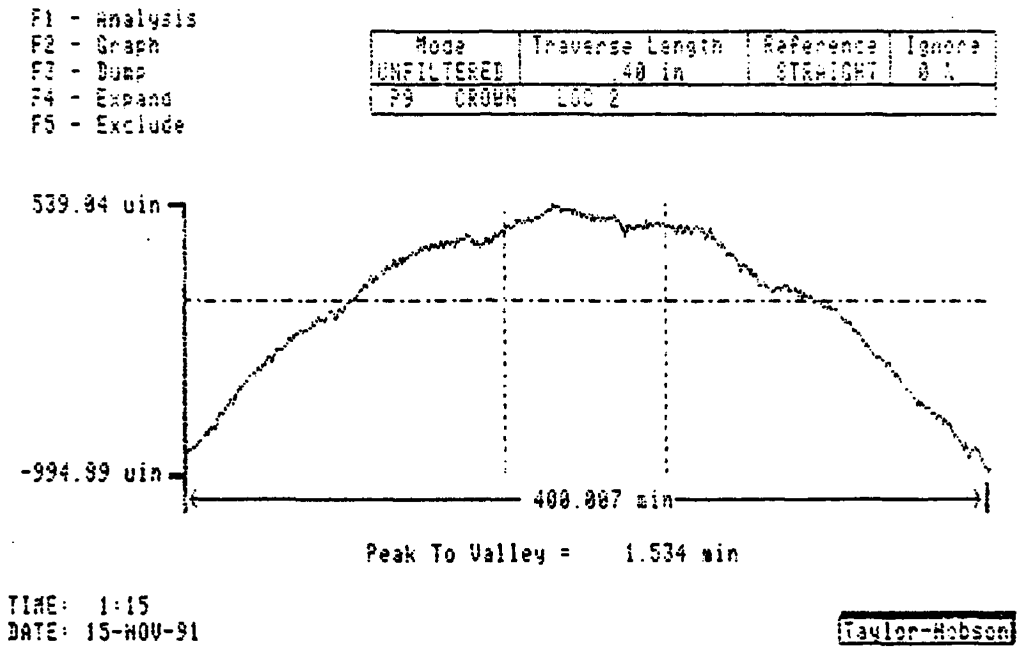


Figure 27a Disc Crown Showing Proper Surface Profile

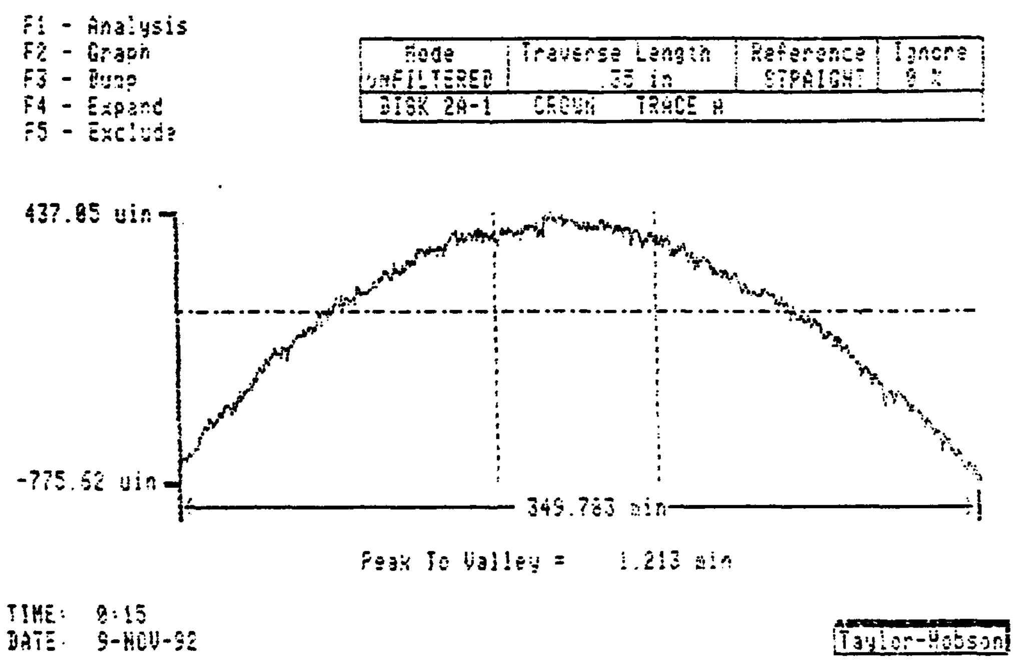


Figure 27b Disc Crown Showing Surface Profile Problem

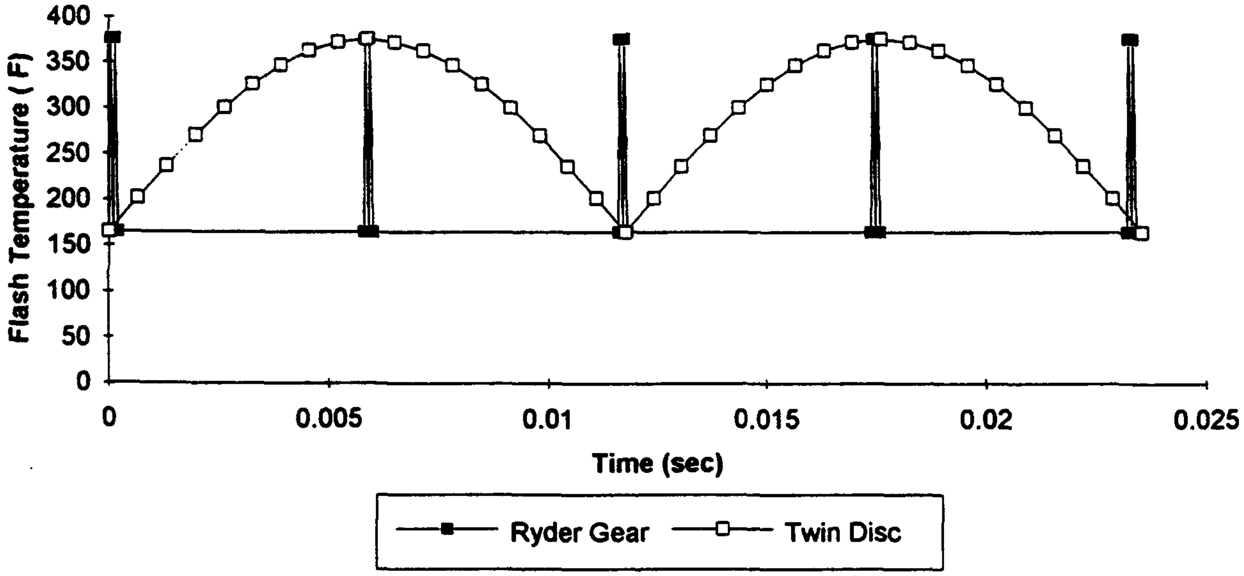


Figure 28 Comparison of Flash Temperature Versus Time for Ryder and Twin Disc Testers

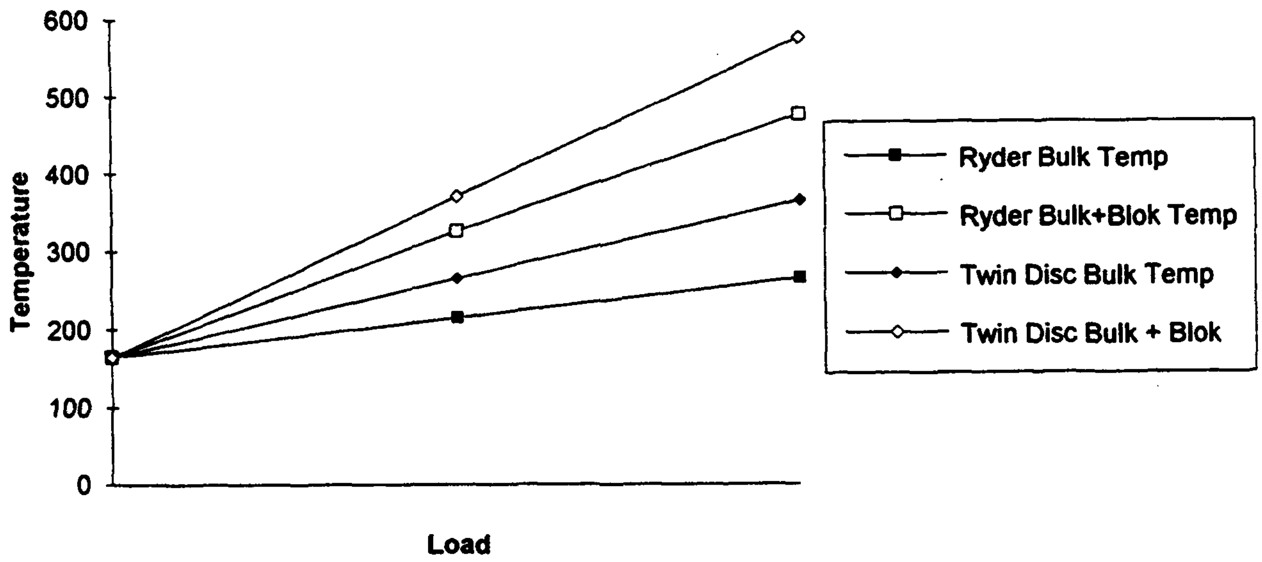


Figure 29 Hypothetical Comparison of Differences in Blok and Bulk Temperatures in Ryder and Twin Disc Testers

7.0 CONCLUSIONS

1. The Twin Disc machine successfully produces scuffing which simulates that found in actual gear tests.

2. Satisfactory correlation was found between Twin Disc and Ryder gear test results. This was especially true for the five Navy lubricants which were ranked identically to the Ryder results. Some experience based expertise may have led to the better consistency of the Navy tests since they were the last tests conducted in the overall program.

3. Control of disc finish and geometry is critical to the test results and may account for the variation in the load capacity values observed. Verifying both the crown profile and finish with a high resolution profilometer is necessary if consistent data are to be obtained. While efforts were made to control the grinding process and the grinding wheel dressing, in retrospect use of a commercial vendor who is used to volume production (such as a bearing manufacturer) might have produced more consistent results. While the crown contour could be controlled reasonably accurately, the finish was more difficult to control. Ordering discs in quantity at the start and conducting all tests with discs ground in one batch may have produced more consistent results.

4. From a metallurgical standpoint regrinding of discs does appear to be feasible although referencing to a standard lubricant may be necessary to compare data on discs manufactured or reground at different times.

5. The failure detection technique planned for the twin disc tester was to measure the contact resistance quality across the EHD film between the two test discs, using an instrument called the ASPERITAC with electrically insulated spindle shafts. The "ASPERITAC" places a small electrical potential across the discs and detects when the electrical resistance across the film was made and broken. For a specific gate time the instrument counts the number of times the measured voltage across the discs pass a preselected threshold and for what percent of time the discs are in actual contact. It was assumed that when the number of contacts was very small and the percent of contact time was high disc contact was pronounced and a scuffing incident was approaching.

Actual experience has shown that particularly with fully formulated lubricants, significant asperity contact can occur at loads considerably below the ultimate scuffing load making contact resistance a poor indicator of the onset of scuffing. While it was not possible to investigate the details of contact resistance performance under this program, there may be significant information available on additive effectiveness in the details of when the contact resistance becomes zero versus when scuffing actually occurs.

6. For evaluation of higher load capacity lubricants such as the Navy helicopter gearbox lubricant the machine should be modified with a higher strength heat dam on the disc mounting spindles to permit evaluation of load capacity at higher loads.

7. Detailed evaluation of the differences between the loads at which the lubricating film becomes conductive as measured by contact resistance and the load at which scuffing occurs may provide insight into lubricant performance in the boundary lubrication regime.

8. In summary, the twin disc tester did provide scuffing load data which compares favorably with Ryder gear test data. The few test anomalies which manifested themselves as either a much higher or a much lower twin disc load do not seem to follow a specific pattern. A coarse disc finish shows both higher and lower scuffing loads when compared to Ryder gear testing and what appears to happen is that a particular lubricant and additive package can perform in unexpected ways depending on what type of test is performed, i.e., gear mesh, or variable slip disc, and how the test specimen surface finish varied.

9. Additional insight into the explanation of the test divergence may be explained by the work of Olver [6] who discusses the thermal and chemical aspects of disc and gear load capacity testing. The detailed understanding of how the bulk and Blok temperatures in the Twin Disc machine compare to those in an actual gear application would make comparison of tester results with anticipated gear performance more meaningful. Conducting a detailed thermal study of the twin disc including both analysis and IR temperature measurements would add significantly to the understanding of this machine.

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APPENDIX A

PARTS LISTS

Twin Disk Gear Tooth Simulator Parts Lists

Component	Part List
Simulator Assembly	PL759J003
Transmission	PL759E049
Frame	PL759J063
Spindle Housing	PL759D064
Pivoting Post	PL759D065
Test Head	PL759E066
Lubricant Supply System	PL759E086

APPENDIX B

DRAWING LIST

Component	Drawing Number	Number of Sheets
Twin Disc Gear Tooth Simulator Assembly	759J003	2
Cap - Load Cylinder	759B012	1
Shim - Stationary Spindle	759B068	1
Coupling	759B073	1
Flange - V-Clamp	759B078	1
V-Clamp	759B079	1
Coupling Guard	759D080	1
Belt Guard	759D081	1
Shim - Motor Support	759B082	1
Motor Isolation Pad	759B083	1
Fitting	759B084	1
Slip Ring	759B091	1
Slip Ring Clamp	759B092	1
Gib	759B095	1
Gib Plate	759B096	1
Gib Plate Insulator	759B097	1
Housing Insulator	759B098	1
Mounting Block-Brush	759B099	1
Insulating Washer	759B100	1
Transmission Assembly	759E049	4
End Cover - Upper Shaft	759C021	1
Seal Housing - Upper Shaft	759C022	1
End Cover - Lower Shaft, Roller Bearing	759C023	1
Seal Housing - Lower Shaft, Roller Bearing	759C024	1
End Cover - Left, Lower Shaft, Ball Bearing	759C025	1
Gasket - Top	759D026	1
Cover - Top	759D027	1
Spanner wrench	759C029	1
Gasket - Bottom	759D036	1
Cover - Bottom	759D037	1
Shaft - Upper	759E042	1
Shaft - Lower, Driver	759E043	2
Shaft - Lower, Driven	759E044	2
Machining - Transmission, Final	759E045	2
Gear Blank, Sinusoidal Gear	759E046	1
Machining - Gear Tooth	759C047	1
Machining - Sinusoidal Gear, Final	759C048	2
Housing - Lower	759E050	2
Housing - Center	759E051	2
Housing - Upper	759E052	1
Spacer - Bearing	759B067	1
End Cover - Right, Lower Shaft, Ball Bearing	759C077	1

Component	Drawing Number	Number of Sheets
Frame Assembly and Components	759J063	4
Cover - Post	759C018	1
Angle	759B075	1
Channel	759B076	1
Spindle Housing Assembly G1 and G2	759D064	1
Housing - Movable	759E004	1
Housing - Stationary	759E005	1
Bearing Retainer - Inner	759C007	1
Bearing Retainer - Outer	759C008	1
Spacer	759B009	1
Washer	759B010	1
Spring Guide	759B011	1
Spindle Shaft - Movable	759D020	1
Spindle Housing Assembly G1 and G2 (continued)		
Heat Dam	759C038	1
Spacer, Sleeve	759C088	1
Assembly, Spindle Shaft - Heat Dam	759D072	1
Pivoting Post Assembly	759D065	1
Post	759D015	1
Stand	759E016	1
Bearing Retainer	759B017	1
Test Head Assembly	759E066	1
End Cap	759B039	1
Cover	759D040	1
Bracket	759E041	1
Housing - Test Head	759E054	1
Housing - Seal	759D055	1
Gasket - Cover	759D056	1
Spacer Ring	759B057	1
Gasket - Seal Housing	759C059	1
Ring - Spacer	759C059	1
Retainer - Seal	759C060	1
Seal - Hydrostatic	759B061	1
Test Specimen	759B062	1
Test Specimen (Nitralloy)	759B102	1
Locking Plate	759B070	1
Locking Device	759B074	1
Seal Ring	759B089	1
Heating Element	759C090	1
Anti-Rotation Pin	759B094	1
High-Temperature Lubricant Supply System	759E086	1
Heat Exchanger	759E087	1
Seal Positioning Cup	SK-B-8005	1
Guide Bushing for Test Head Alignment	SK-B-8006	1

APPENDIX C

HAZARD ANALYSIS REPORT - TWIN DISC GEAR TOOTH SIMULATOR
(Contract F33615-86-C-2622)

HAZARD ANALYSIS REPORT - TWIN DISC GEAR TOOTH SIMULATOR

An analysis to identify and control hazards and to determine safety requirements for the Twin Disc Gear Tooth Simulator and its subassemblies was performed. The results of the analysis, presented herein, include the means for preventing injury to operating and maintenance personnel, for preventing simulator damage, and for environmental protection. The simulator as a whole and each of the following subassemblies were analyzed separately:

- Test Head
- Transmission
- Spindles
- Transmission Lubrication System
- Test Lubricant Supply System.

The hazard analysis provides a brief description of the simulator and each subassembly, describes the hazards, identifies the means by which a hazard can be detected and/or the conditions under which a hazard may occur, classifies the level of risk to personnel and the simulator, and recommends actions to be taken to prevent hazard occurrence.

TWIN DISC GEAR TOOTH SIMULATOR

The Twin Disk Gear Tooth Simulator is a test machine that evaluates load capacity of lubricants and gear material performance. The machine uses two discs that roll together, under load, with a degree of sliding that varies from zero to a positive and negative maximum value during each revolution. A radial load is applied to a test disc as it rotates against a second disc, which is also rotating. A special geared transmission provides cyclic variations in the rotational speed of the two discs so that both rolling and sliding friction are experienced by both test discs. Hot oil is injected between the test discs for lubrication purposes.

Hazard Description

- Fire
- Noxious vapors
- Uncontaminated lubricant
- High nose levels
- Rotating machinery components
- Pressurized air
- Pressurized hot oil
- High-temperature components
- Electrical shock
- Unlubricated machine start-up
- Machine start-up under load.

Hazard Identification

- Seeing or smelling the lubricant or lubricant by-product liquids or vapors
- Seeing or smelling smoke or fire
- Hearing excessive noise

- Operating the simulator under the following conditions
  - Without guards in position
  - Without covers properly installed
  - When electrical connections are uncovered
  - With missing or damaged insulation
  - With missing or disconnected safety interlocks.

Risk Assessment

- The risk to personnel could be classified as critical.
- The risk to the simulator could be classified as critical.

Recommended Action

- At the installation site, provide industrial forced-draft ventilation and noise protection. A separate cell for the simulator is preferred.
- Install all electrical systems per NFPA electrical code
- Provide safety interlocks that prevent drive motor actuation before the activation of appropriate lubricant supplies.
- Provide safety interlock that prevent the imposition of test load before the activation of the drive motor and test lubricant supply.
- Place type B/C and D fire extinguishers at the test site in an area where they will be easily accessible under emergency conditions.
- Train all test site personnel in the safe handling of special lubricants and in general simulator operation.
- Apply safe color code ANSI Z53.1-1967 for marking physical hazards to appropriate components.
- In the simulator control area, provide a material safety data sheet for each lubricant used in the simulator.
- Prior to conducting any maintenance, disconnect all electrical systems.
- Periodically inspect electrical, air, water, and oil lines. Replace defective lines.
- Wear eye and ear protection when working near the simulator while it is operating. At no time, however, should you be close to the simulator while it is operating at full load and speed.
- Before and during simulator operation, make sure that all electrical covers and machinery guards and covers are in place.
- Immediately remove clothing that has been splashed or soaked with lubricating oils. Launder clothing before reuse.
- Wash skin that has come in contact with lubricating oils with soap and water.
- Wipe up spills promptly to prevent the possibility of slips and falls.

TEST HEAD

The test head is a sealed chamber with penetrations for shafting, heaters, lubricant supply, and instrumentation. This subassembly includes two test discs, two shafts, two seals, and several types of instrumentation.

Hazard Description

- Fire
- Noxious vapors
- Uncontained lubricant
- High noise levels
- Rotating machinery components
- Pressurized air
- Pressurized hot oil
- High-temperature components
- Electrical shock
- Unlubricated machine start-up
- Machine start-up under load.

Hazard Identification

- Seeing or smelling the lubricant or lubricant by-product liquids or vapors
- Seeing or smelling smoke or fire
- Hearing excessive noise
- Operating the simulator under the following conditions
  - Without guards in position
  - Without covers properly installed
  - When electrical connections are uncovered
  - With missing or damaged insulation
  - With missing or disconnected safety interlocks.

Risk Assessment

- The risk to personnel could be classified as critical.
- The risk to the simulator could be classified as critical.

Recommended Action

- Before and during simulator operation, make sure all electrical covers and machinery guards and covers are in place.
- Make sure safety color code ANSI Z53.1967 has been applied as appropriate.
- Periodically inspect lubrication supply and drain lines. Replace defective lines.
- Use adequate forced-draft ventilation.
- Torque test disc attachment fasteners to recommended value; follow start-up instructions.

- Prior to simulator operation, inspect the buffer gas system including system vents; follow start-up instructions.
- Make sure a material safety data sheet for each lubricant used in the simulator is available in the simulator control area.
- Follow the instructions for the safe handling of any lubricants used in the simulator.
- Periodically examine all electrical systems for loose or missing covers, damaged wires, and disconnected or missing safety interlocks.
- Wear eye and ear protection when working near the simulator while it is operating.

TRANSMISSION

The transmission is a gearbox assembly that transforms a constant, single shaft input speed to a non uniform cyclic output speed on each of two output shafts. the output shafts of the transmission ultimately drive the twin disc test discs.

Hazard Description

- Unlubricated transmission operation
- Rotating machinery components
- Oil spray.

Hazard Identification

- Starting the transmission under the following conditions:
  - Without lubricant pressure
  - With guards or covers missing.

Risk Assessment

- The risk to personnel could be classified as critical.
- The risk to the simulator could be classified as critical.

Recommended Action

- Make sure safety color code ANSI Z53.1-1967 has been applied to the input and both output shaft couplings of the transmission.
- Make sure that a safety interlock that prevents drive motor actuation without adequate lubricant supply pressure has been provided and is operational.
- Follow operating instructions.
- Before simulator operation, make sure that transmission covers or adequate substitutes (such as transparent covers that allow you to see the transmission during maintenance) are in place.

SPINDLES

Two ball bearing spindles are included in the simulator design. These spindles transfer the rotary motion of the transmission output shafts to the test disc and have the capability of imposing high radial test loads.

Hazard Description

- Overheating of spindle bearings.

Hazard Identification

- Seeing that spindle temperatures are above established normal range.

Risk Assessment

- The risk to personnel could be classified as non critical.
- The risk to the simulator could be classified as critical.

Recommended Action

- Monitor spindle bearing temperatures with fixed thermocouples. If you see excessively high readings, shut down the simulator and investigate the heat source.

TRANSMISSION LUBRICATION SYSTEM

The transmission lubrication system is a self-containing hydraulic pumping unit with the sole purpose of supplying pressurized lubrication to the simulator transmission.

Hazard Description

- Rotating machinery components
- Spilled oil
- Oil spray.

Hazard Identification

- Operating the lubrication system without coupling guard in position
- Seeing oil spray and/or spilled oil
- Hearing excessive noise
- Seeing evidence of above-normal temperatures.

Risk Assessment

- The risk to operating personnel could be classified as moderate.
- The risk to the simulator could be classified as critical.

Recommended Action

- Before starting the lubrication system, make sure all guards are in place.

- Periodically check oil level in lubrication sump for sufficient lubricant quantity.
- Immediately remove clothing that has been splashed or soaked with lubricating oils. Launder clothing before reuse.
- Wash skin that has come in contact with lubricating oils with soap and water.
- Wipe up spills promptly to prevent the possibility of slips and falls.

TEST LUBRICANT SUPPLY SYSTEM

The transmission lubrication system is a self-containing hydraulic pumping unit with the sole purpose of supplying pressurized lubrication to the simulator transmission.

Hazard Description

- Rotating machinery components
- Spilled oil
- Oil spray.

Hazard Identification

- Operating the lubrication system without coupling guard in position
- Seeing oil spray and/or spilled oil
- Hearing excessive noise
- Seeing evidence of above-normal temperatures.

Risk Assessment

- The risk to operating personnel could be classified as moderate.
- The risk to the simulator could be classified as critical.

Recommended Action

- Before starting the lubrication system, make sure all guards are in place.
- Periodically check oil level in lubrication sump for sufficient lubricant quantity.
- Immediately remove clothing that has been splashed or soaked with lubricating oils. Launder clothing before reuse.
- Wash skin that has come in contact with lubricating oils with soap and water.
- Wipe up spills promptly to prevent the possibility of slips and falls.

TEST LUBRICANT SUPPLY SYSTEM

The test lubricant supply system is a self-contained hydraulic pumping unit, specifically designed to supply specialized and exotic test lubricants under pressure and at high temperatures to the simulator test head.

Hazard Description

- Fire
- Noxious vapors
- Pressurized hot oil spray
- Spilled oil
- High-temperature components
- Electrical shock
- Unlubricated pump start-up.

Hazard Identification

- Seeing or smelling the lubricant or lubricant by-product liquids or vapors
- Seeing or smelling smoke or fire
- Operating the test lubricant supply system under the following conditions:
  - With defective or missing insulation
  - Without electrical component covers installed
  - With an empty oil sump.

Risk Assessment

- The risk to personnel could be classified as critical.
- The risk to the simulator could be classified as moderate.

Recommended Action

- Make sure all electrical equipment has been installed per NPFA electrical code.
- Make sure that type B/C and D fire extinguishers are easily accessible and adequately charged.
- Make sure that a material safety data sheet for each lubricant used in the simulator is available in the simulator control area.
- Prior to simulator start-up, make sure all electrical covers are in place.
- Do not use certain combinations of materials and lubricants for high-temperature testing, i.e., aluminum, titanium alloys, or magnesium with Krytox.
- Use adequate forced-draft ventilation.
- Prior to conducting any maintenance, disconnect all electrical systems.
- Periodically check lubricant level in sump for sufficient lubricant quantity.
- Immediately remove clothing that has been splashed or soaked with lubricating oils. Launder clothing before reuse.
- Wash skin that has come in contact with lubricating oils with soap and water.
- Wipe up spills promptly to prevent the possibility of slips and falls.

APPENDIX D

OPERATING INSTRUCTIONS

1.0 INTRODUCTION

The operating instructions tell you how to:

- Clean and install the discs prior to each testing cycle
- Perform the production testing
- Change the test oil.

The following drawings are referenced:

- 759E066 - Test Head Assembly
- 759E086 - High-Temperature Lubricant Supply System.

2.0 DISC CLEANING, INSTALLATION, AND REMOVAL

> ***CAUTION***
>
> *Handle discs carefully. You can damage the disc test surfaces (outer diameter) with scratches, dents, or fingerprints if you do not follow proper handling procedures.*

> ***CAUTION***
>
> *For test purposes, use only discs that meet quality assurance specifications. Using discs that do not meet specifications may invalidate test results.*

2.1 Cleaning

1. Prior to each test cycle, clean the test discs for 20 min in an ultrasonic cleaner charged with Stoddard solvent.
2. Prior to installation, wipe all disc surfaces with a lint-free cloth dampened with Stoddard solvent. You may use lint-free gloves to prevent placing fingerprints on the crowned disc OD.

2.2 Installation

1. Preheat discs for 30 min in an air oven set to 350°F.
2. Clean spindle shaft mounting surfaces (diameter and shculder) with a lint-free cloth dampened with Stoddard solvent.
3. With the aid of puller bolts in stalled in two puller holes in each disc, assemble discs to spindle shaft ends. Seat the discs firmly against the spindle shoulder. Refer to Drawing 759E066 for proper disc positioning. Align circumferential spindle and spindle housing marks and place the discs with their identification marks at top dead center (TDC). If installing reworked discs, place the disc identification number at 90° past TDC.
4. Install Ringfedder clamps, Ringfedder clamp end caps, and appropriate nuts and nut retainers. Torque nuts to 150 in.-lb. Refer to Drawing 759E066 for proper component positioning.
5. Install appropriate test head cover as follows, referring to Drawing 759E066 for proper cover positioning.
   a. For test oil temperatures below 200°F, install polycarbonate test head cover. Torque cover bolts firmly.
   b. For test oil temperatures of 200°F and higher, install Inconel test head cover. Torque cover bolts to 30 in.-lb.

The simulator is ready for production testing.

2.3 Removal

1. Disconnect all power to simulator.
2. Remove test head cover.
3. Remove Ringfedder clamp end caps.
4. Install disc puller with center protector plug.
5. Install (hand tight) three 1/4 20 × 2-1/4 in. long puller bolts. (Be sure to maintain puller squareness to spindle shaft.)
6. Turn puller center bolt until disc is free of spindle.
7. Remove all Ringfedder clamp parts.

3.0 PRODUCTION TESTING

3.1 Machine Setup

1. Connect power to simulator.
2. Start transmission lubricant pump.
3. Make sure relief valve is set at 40 to 45 psig.
4. Turn on seal gas supply and set pressure at 40 to 50 psig.
5. Make sure the load condition is at 0 lb.
6. Start drive motor.
7. Set speed at 350 to 450 rpm.

3.2 Instrumentation Setup

1. Make sure instrument power is on.
2. Set ELECTAC to the following settings:
   - Contact voltage: 0.1 V
   - Discriminator voltage: 0.05 V
   - Center resistance: 1 kΩ
   - Integrator gain: 1
   - Clock: 1 sec

3.3 Test Oil Start-Up

1. Make sure the loop shut-off valve (Item 7, Drawing 759E086) is open.
2. Make sure the test oil shut-off valve (Item 19, Drawing 759E086) is open.
3. Make sure the pump bypass valve (Item 15, Drawing 759E086) is fully open.
4. Adjust lubricant delivery needle valve (Item 25, Drawing 759E086) fully open.
5. Check sump level. Add oil as required. The lubricant supply pump requires a test lubricant viscosity less than 200 cs for safe operation. Any test oil not meeting this requirement at room temperature must be heated to a minimum temperature that will provide this viscosity. To accomplish this, the suction line heater between the sump and the pump and the sump heater should be energized to a partial power level commensurate with the required minimum oil temperature. When the desired sump temperature is reached, proceed to Step 6.
6. Start test oil pump.

> **NOTE**
>
> *Operating the simulator for a low number of test cycles should not degrade the test oil. If extended testing is conducted, check oil for viscosity and neutralization number. Replace oil when degradation is noted (see Section 4.0).*

3.4 Testing

1. Adjust load to 0 lb.
2. Set lubricant delivery valve (Item 25, Drawing 759E086) fully open.
3. Adjust lubricant temperature to desired test temperature (±5°F). Adjustment of test oil temperature is accomplished with the sump heater (Item 27, Drawing 759E086) and the line heaters (Items 28, 29, and 30, Drawing 759E086). At oil temperatures below 200°F, only the sump heater may be required. At oil temperatures above 200°F, all heaters may be needed. Do not exceed a sump oil temperature reading of 450 to 480°F and line temperature readings of 750°F.

> ***NOTE***
>
> *Each test oil will require different heater settings, some pretest evaluation may be helpful.*

4. Adjust speed to 2500 rpm.

> ***NOTE***
>
> *Transmission oil temperature must be above 93° F to promote proper drainage at higher speeds.*

5. Adjust load to obtain a minimum ASPERITAC (ELECTAC) reading of approximately 75 to 85%. Adjust load no higher than the level at which a 100% reading is first obtained.
6. Run simulator for 2 min and record load and reading on data sheet*.
7. Adjust load upward to obtain an ASPERITAC (ELECTAC) reading of approximately 85 to 95%. Run load for 2 min and record data.

> ***NOTE***
>
> *Under certain conditions, Step 7 will immediately produce a reading of 100%. If this occurs, go to Step 9. Early failure detection will occur when the contact count readings start to increase at a fixed or increasing load.*

8. Repeat Step 7 until load reading is consistently at or above 95%.
9. When reading remains above 95 to 100% for more than 2 min, remove the load, stop the simulator, and examine the discs.
10. Proceed as follows if discs are:
    a. Scuffed - terminate testing.
    b. Not scuffed - resume speed and increment load by 30 to 60 lb. Run for 2 min. Remove the load, stop the simulator, and examine the discs. If discs are not scuffed repeat Step 10b: if scuffed, terminate test.

---

* A data sheet that you may copy for this purpose is provided at the end of this section.

TWIN DISC LUBRICANT EVALUATION TEST

ASPERITAC Settings:

Contact Volts: ______________________ Date: ______________________

Descrim Volts: ______________________ Sheet _____ of _____

Center Resistance: ______________________ Data Point: ______________________

Integrator Gain: ______________________ Test Oil: ______________________

	Identification	Crown Radius	0° $\sqrt{A}$	0° $\sqrt{C}$	90° $\sqrt{A}$	90° $\sqrt{C}$
Left Disc						
Right Disc						

Time	Speed (rpm)	Load (lb)	Temperature (°F)	Percent	Count
	2500				
	2500				
	2500				
	2500				
	2500				
	2500				
	2500				
	2500				
	2500				
	2500				
	2500				
	2500				
	2500				
	2500				
	2500				
	2500				
	2500				
	2500				
	2500				
	2500				

4.0 CHANGING TEST OIL

When a change of test oil is required, proceed as follows. (See Drawing 759E086 for all items listed below.)

1. Drain all test oil from system through drain plug in suction filter (Item 8).
2. Remove test head cover and test discs, if present.
3. Remove all filter elements (Items 8 and 23).
4. Remove heat exchanger and hose as an assembly (Items 47 and 49), thermowell (Item 48) and sump flange assembly (Item 1).
5. Clean all components in a suitable cleaning tank and then flush with clean Stoddard solvent.
6. Wipe inside surface of sump and test head with lint-free cloth dampened with clean Stoddard solvent.
7. Reinstall all components removed in Step 4, install new gaskets (gasket identification can be found on Drawings 759E086 and 759E066).
8. Charge test oil system with 4 liters of clean Stoddard solvent.
9. Install test head cover.
10. Circulate solvent in test oil loop for 30 min:
    a. Fully open bypass valve (Item 15).
    b. Fully open delivery valve (Item 25).
11. Drain solvent and discard.
12. Repeat Steps 8 to 11 one more time, and then proceed to Step 10.
13. Fill sump with 2 liters of test oil.
14. Install test head cover.
15. Circulate test oil through all piping and the test head for 15 min.
16. Drain oil and discard.
17. Reinstall filter elements removed in Step 3.
18. Refill test oil system with 4 liters of new test oil.

The simulator is now ready for disc installation and production testing.

APPENDIX E

MAINTENANCE INSTRUCTIONS

1.0 INTRODUCTION

The twin disc simulator provides a trouble-free design that requires minimal maintenance. However, in the event that the machine does require adjustment or repair, the maintenance instructions tell you how to perform the necessary procedures on the:

- Drive belt
- Seals
- Transmission
- Spindles.

The following drawings are referenced:

- 759J003, Twin Disc Gear Tooth Simulator Assembly
- 759E043, Shaft - Lower, Driver
- 759E044, Shaft - Lower, Driven
- 759E049, Transmission Assembly
  - 759D064G1
  - 759D064G2
- 759E066, Test Head Assembly
- SK-B-8805, Seal Positioning Cup

2.0 DRIVE BELT

2.1 Tension Adjustment

- Remove belt guard.
- Loosen four bolts (Item 49, Drawing 759J003).
- Shift drive motor position by adjusting four motor mount screws (Item 30, Drawing 759J003).
- Check pulley alignment with straight edge held against both pulley flanges. Adjust alignment as required.
- Tighten four bolts (Item 49, Drawing 759J003).
- Replace belt guard.

2.2 Belt Replacement

- Remove belt guard.
- Loosen four bolts (Item 49, Drawing 759J003).
- Shift drive motor position sufficiently to remove belt by adjusting four motor mount screws (Item 30, Drawing 759 J003).
- Replace belt and adjust belt tension per Section 2.1, Steps 3 and 4.
- Tighten four bolts (Item 49, Drawing 759J003).
- Replace belt guard.

3.0 SEAL REPLACEMENT

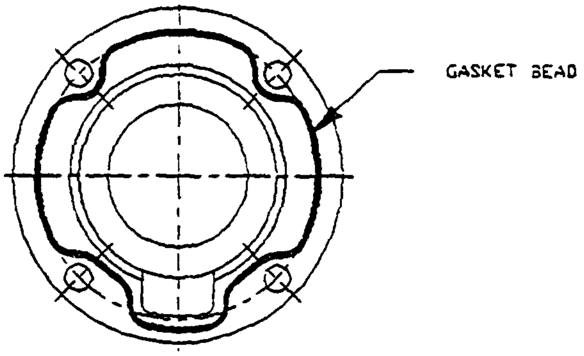
3.1 Test Head Seals

1. If discs are installed, remove them (see Part I, Operating Instructions, Section 2.3).
2. Remove cooler clamp (Item 17, Drawing 759J003).
3. Remove test head bolts and taper dowels (Items 35 and 23, respectively, Drawing 759E066).
4. Disconnect test oil supply line.
5. Remove heater power wires. If installed, make sure heating elements will clear test lube oil system when the test head is removed.
6. Disconnect thermocouples.
7. Carefully slide test head housing off guide pins.
8. Place test head on clean surface, gasket side down.
9. Remove seal retainers (Item 11, Drawing 759E066).
10. Replace seal components.
11. Measure width of seal spacer against width of seal. Adjust spacer to provide a 0.001 to 0.002 in. clearance between the spacer and the carbon seal ring.
12. Replace seal retainer(s).
13. Torque bolts (Item 34, Drawing 759E066) to 20 in.-lb.
14. Wipe test head mounting surfaces with clean lint-free cloth.
15. Install seal positioning cups (SK-B-8805) onto spindle shafts.
16. Carefully position test head on alignment dowels and push into position.
17. Install and hand-tighten the four test head bolts.
18. Install two tapered dowels.
19. Torque test head bolts to 50 ft-lb.
20. Reconnect oil supply line.
21. Reconnect thermocouples.
22. Install cooler clamp (Item 17, Drawing 759J003) with new gasket (Item 35, Drawing 759J003).

3.2 Lower Shaft Transmission Seal

1. Remove belt (see Section 2.2).
2. Remove drive shaft pulley.
3. Disconnect oil feed line from fitting on seal housing.
4. Unbolt and remove seal housing (Item 20, Drawing 759E049).
5. Clean out old gasket material from seal housing and transmission case.
6. Replace seal per manufacturer's instructions.

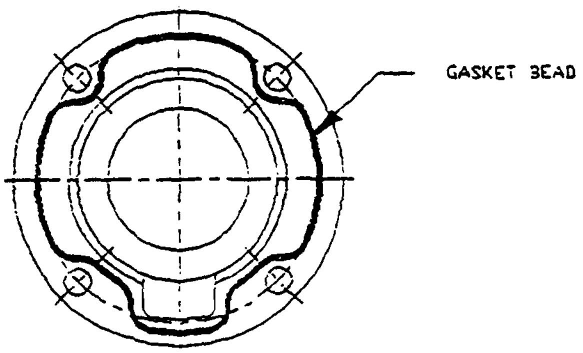
7. Apply "Loctite" primer H to the seal housing and transmission case mating surfaces.
8. Apply a 1/16-in. diameter bead of "Loctite" gasket eliminator No. 515 to the seal hosing as shown below.



9. Install seal housing with drain relief down.
10. Torque bolts to 50 in.-lb.
11. Reconnect oil feed line to fitting on seal housing.
12. Replace drive shaft pulley.
13. Replace and adjust drive belt (see Section 2.0).

3.3 Upper Shaft Transmission Seals

1. Remove transmission (see Section 4.1).
2. Remove transmission output shaft coupling hubs with a suitable puller (see Section 4.3.1, Step 2). Protect the shaft center during this operation. Maintain hub/shaft identification.
3. Unbolt and remove seal housing(s) (Item 18, Drawing 759E049).
4. Clean out old gasket material from seal housing and transmission case.
5. Replace seals per manufacturer's instructions.
6. Apply "Loctite" Primer H to the seal housings and tran·smission case mating surfaces.
7. Apply 1/16-in. diameter bead of "Loctite" gasket eliminator No. 515 to the seal housing as shown below.



8. Install seal housings with drain reliefs down.
9. Torque bolts to 50 in.-lb.
10. Reinstall coupling hubs, keeping proper hub/shaft installation. Install per Section 4.6.3, Steps 4 and 5.
11. Reinstall transmission (see Section 4.6.5).
12. Install drive belt pulley on transmission shaft.
13. Replace and adjust drive belt (see Section 2.0).

4.0 TRANSMISSION

4.1 Transmission Removal

1. Loosen drive belt and remove drive belt pulley from transmission shaft.
2. Disconnect output couplings at upper transmission shafts.
3. Disconnect oil supply line.
4. Disconnect oil drain line.
5. Remove taper dowel pins.
6. Unbolt transmission from test frame.

4.2 Transmission Disassembly

1. Remove all oil supply tubing.
2. Remove test head cover.
3. Remove all shaft end covers and seal housings. On coupling end of output shafts, remove bolts only.
4. Drive 3/8-in. dowel pins out of the transmission housing (two dowels in each of both intermediate housing flanges).
5. Unbolt and remove upper transmission housing. Take care not to harm mating surfaces (self-gasketing) of housing components.
6. Remove upper shafts with their bearings.
7. Unbolt and remove intermediate transmission housing.
8. Remove lower shafts with their bearings.

4.3 Bearing Removal

4.3.1 Needle Bearings

1. Remove appropriate snap ring(s).
2. On shaft with couplings, remove couplings using one of the following methods: 1) on a hydraulic press, support the coupling hub and press the shaft out, or 2) use a screw-type wheel puller. Protect the shaft center while removing couplings.
3. Remove bearing sleeves with an appropriate puller or by grinding. Protect shaft center when using a puller.

4.3.2 Ball Bearings

1. Remove bearing retaining nuts and locking washers.
2. Remove bearings with suitable puller. Protect shaft center during this operation.

4.4 Seal Removal

1. Remove seal housings

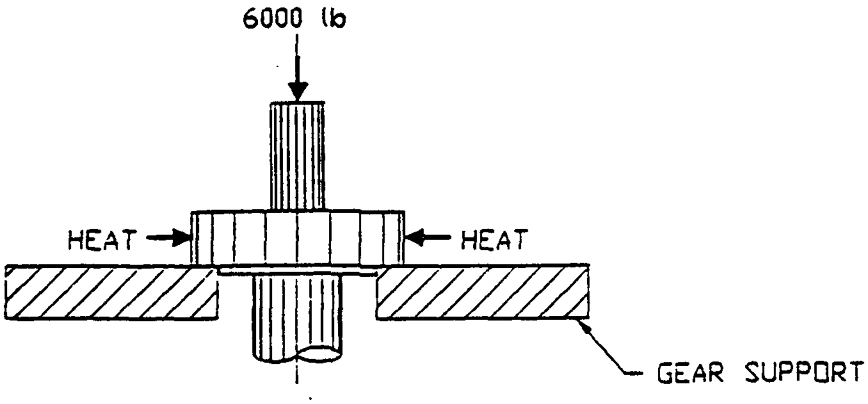
> **NOTE**
>
> *To remove housings on upper transmission shafts, you must first remove the coupling hubs (see Section 4.3.1, Step 2).*

4.5 Gear Removal

1. Support gear/shaft assembly in arbor press. Position the support so that it will not interfere with gear mounting polygon on the shaft. Protect shaft center during this operation.
2. Apply 6000-lb load on shaft.
3. Using dispersion-type nozzles, apply high heat (oxyacetylene) to outer diameter of gear.
4. Maintain heat and load until shaft clears gear.

> **CAUTION**
>
> *All gears are in matched sets and are identified as either driver or driven gears. Make sure the gears stay together as these matched sets. (Drive gears with the larger polygons are mounted on the lower shafts.)*

The figure below illustrates the required setup for gear removal.

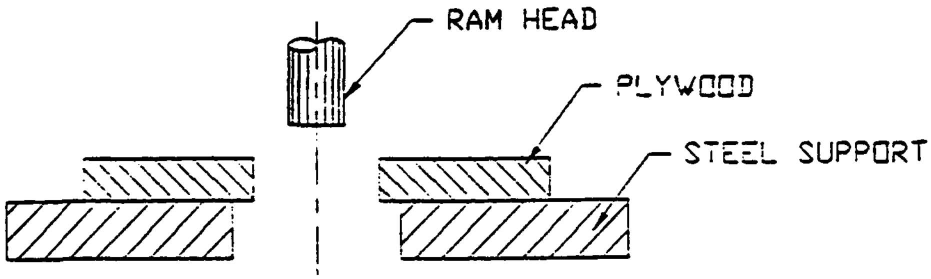


4.6 Transmission Assembly

4.6.1 Gear Installation

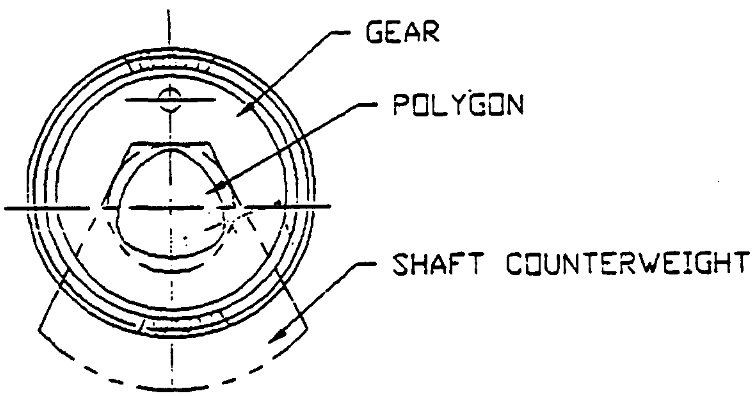
> ***CAUTION***
>
> *All gears are in matched sets and are identified as either driver or driven gears. Make sure the gears stay together as these matched sets. (Drive gears with the larger polygons are mounted on the lower shafts.) During installation, make sure the gears all face the same direction. Operation with gears that are mismatched or facing the wrong direction can damage the simulator.*

1. Place gears in oven set at 300°F and soak for at least 2 hr.
2. Prepare assembly area as shown below.



> ***NOTE***
>
> *Bore plywood to clear all shaft diameters.*

3. Immerse appropriate shaft end in LN2 at least up to the gear seating flange.
4. Keep shaft end immersed in LN2 until bubbling stops.
5. In rapid order:
   a. Place hot gear onto plywood support with appropriate face up.
   b. Insert appropriate shaft end into polygon in gear with counterweight as shown below.



   c. Seat shaft polygon against shoulder.

6. Apply axial pressure (up to 1000 lb) with hydraulic ram to ensure proper seating.

4.6.2 Transmission Shaft Balancing

1. Loosen flange bolts on lower shafts.
2. Replace 1/8-in. diameter dowels.
3. Tighten bolts to 14 ft-lb.
4. Remove dowels.
5. Balance each transmission shaft assembly to the following specification:
   Upper Shafts: 0.013 to 0.015 in.-oz/plane
   Lower Shaft (Drawing 759E043): 0.032 in.-oz/plane
   Lower Shaft (Drawing 759E044): 0.029 in.-oz/plane.
6. Make all corrections for shaft balance on the counterweights when possible. Use one of the following methods:
   a. Drill new holes.
   b. Tap existing holes and install set screws.

> **NOTE**
>
> *If you cannot make corrections on the counterweights, you may remove weight from the gears, providing that you limit the removed material to an area between the shaft flange and 1/4 in. below the gear tooth dedendum. Remove material from both sides of the gear in equal proportions when possible. Blend and polish removal area.*

4.6.3 Gear Lapping

Before completing the final assembly of the transmission, lap the replacement gears according to the following procedure:

1. Install shaft bearings.
2. Install needle bearing races:
   a. Heat bearing races to 250°F.
   b. Slide bearing sleeves on shaft.
   c. Install snap rings.
3. Install ball bearings onto the lower shafts on an arbor press, protect shaft ends, and push bearings only by inner race. Install locknuts and washers.
4. Heat coupling in air oven set at 350°F for 1 hr.
5. Install coupling hubs on upper shafts so that hubs are flush with the shaft ends.

> **NOTE**
>
> *Both couplings are marked for identification. Once you install the coupling hubs on the upper transmission shafts, the location of these shafts in the transmission is fixed.*

4.6.4 Transmission Assembly for Lapping

1. Assemble lower transmission housing and cover.
2. Torque cap screws to 50 in.-lb.
3. Lubricate all lower shaft bearings and gears with Mobil 630 oil.
4. Install lower transmission shafts. Be sure to align drive gear timing marks.
5. Install intermediate transmission housing.
6. Hand tighten 3/8-24 housing bolts.
7. Install 3/8-in. diameter dowels.
8. Tighten 3/8-24 housing bolts to 45 ft-lb.

> **NOTE**
>
> *Transmission case members are match-marked for proper orientation.*

9. Lubricate all upper shaft bearings and gears with Mobil 630 oil.
10. Install upper transmission shafting. Be sure to maintain gear identification and timing mark alignments with lower shafting.
11. Install upper transmission housing.
12. Hand tighten 3/8-24 housing bolts to 45 ft-lb.
13. Install 3/8-in. diameter dowels.
14. Tighten 3/8-24 housing bolts to 45 ft-lb.
15. Rotate the transmission shafts slowly until the adjacent innermost upper shaft sinusoidal gears interfere with each other. This interference is expected.
16. Carefully remove material from the addendum of the interfering gear teeth by hand with a medium-grid stone until a 0.005-in. feeler gage passes freely between the teeth.
17. Clean away the grinding debris. If necessary, reverse the above assembly procedure to clean individual components.
18. Remove the 1/8-in. diameter dowel from the torquing flanges.
19. Loosen the flange bolts on the lower shafts.
20. With the tools provided, apply a 10 to 12 ft-lb torque to lower shaft segments.

21. Tighten the flange bolts to 90 in.-lb.

> ***NOTE***
>
> *There is no preferred torquing direction; however, once you establish a torquing direction, you must maintain it for all subsequent operations. There may be a slight variation in backlash prior to torquing. The minimum backlash location should be the position at which the torquing is performed.*

22. Prepare the gear lapping compound by mixing equal parts of Andox C grease with "TimeSaver"* lapping compound 111N (fine). Paint all the sinusoidal gear surfaces with liberal amounts of the mixture.

> ***NOTE***
>
> *Use thin rubber aprons cut to fit around the four transmission shafts to inhibit the migration of lapping compound into the shaft support bearings.*

23. Install the transmission on the test stand reversing the removal procedure (Section 4.1). You do not need to connect the supply or drain lines at this time. Since the upper transmission shafts do not have thrust bearings, connect the upper shaft couplings.
24. Run the transmission at 300 rpm for 30 min.
25. Check several sinusoidal gear teeth for finish. If teeth do not appear lapped, continue lapping.
26. Run the transmission in 15 min intervals, checking the teeth after each interval, until gear teeth look lapped but do not have a well-defined lateral line at the pitch line location. Too little lapping is better than too much. Add Mobil 630 oil to bearing feed holes periodically during this procedure.
27. Remove transmission from test stand.
28. Disassemble transmission, except do not remove ball bearings or needle bearing sleeves unless they are damaged by the lapping compound.
29. Clean all transmission components until you remove all traces of lapping compound.
30. Re-oil surfaces and bearings with Mobil 630.
31. Remove coupling hubs from transmission's upper shafts.

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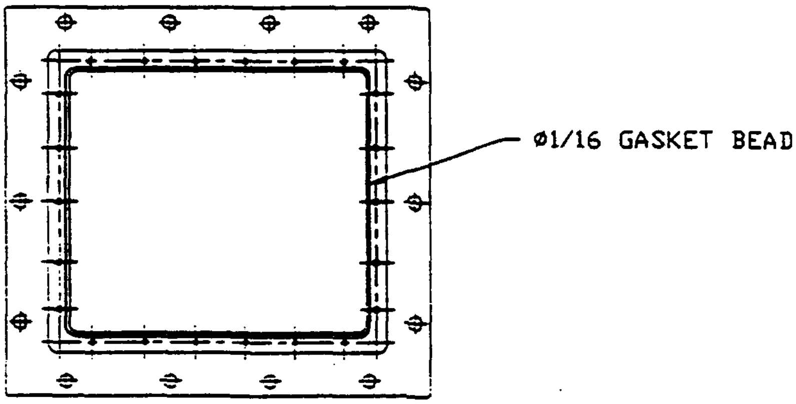
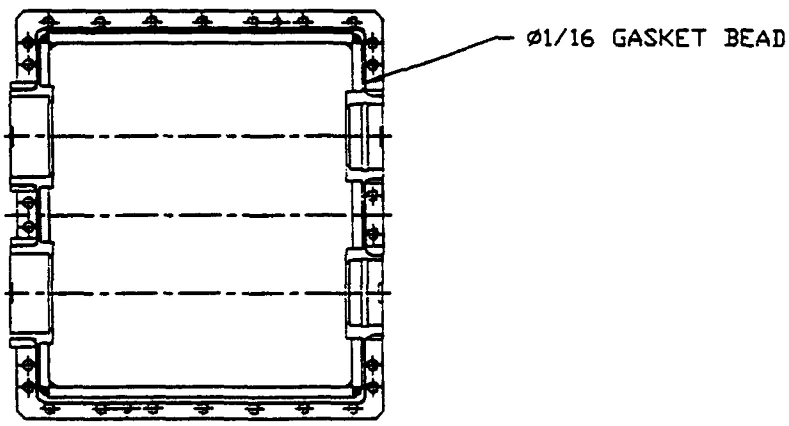
* TimeSaver Products Co., Franklin Park, IL.

4.6.5 Final Transmission Assembly

1. Install ball and needle bearings, as required.
2. Lubricate all lower shaft bearings and gears with Mobil 630 oil.
3. Reinstall transmission bottom cover if removed earlier. Apply "Loctite" Primer H to mating surfaces on both bottom cover and transmission lower housing. Apply a 1/16-in. diameter bead of "Loctite" gasket eliminator No. 515 to lower housing bottom cover seat as shown below.

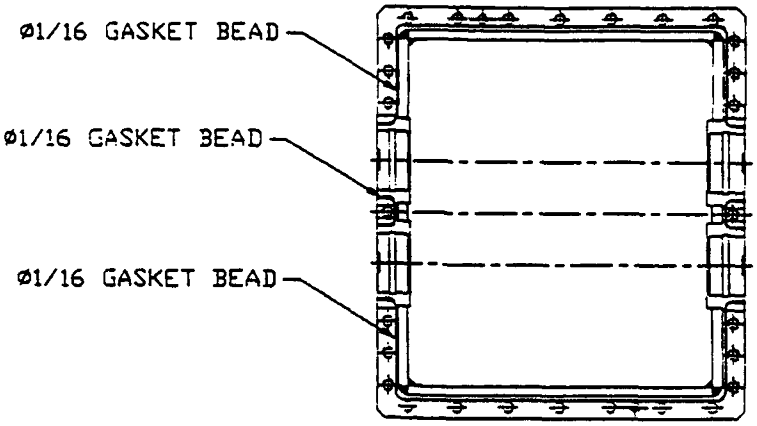


4. Apply "Loctite" Primer H to the lower housing flange and to the flanges of both the intermediate and upper housings.
5. Install lower shafts. Be sure to align drive gear timing marks. Apply a 1/16-in. diameter bead of "Loctite" gasket eliminator No. 515 to the lower housing as shown below.

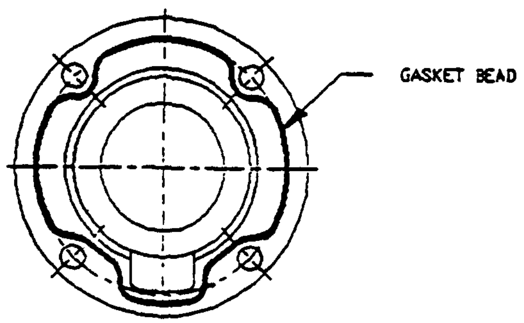


6. Install intermediate transmission housing. Hand tighten 3/8-24 bolts. Install 3/8-in. diameter dowels. Tighten 3/8-24 bolts to 45 ft-lb.

7. Lubricate all upper shaft bearings and gears with Mobil 630 oil.
8. Install upper transmission shafting. Be sure to maintain gear identification and timing mark alignments with lower shafting.
9. Apply a 1/16-in. diameter bead of "Loctite" gasket eliminator No. 515 to the intermediate housing as shown below.



10. Install upper transmission housing. Hand tighten 3/8-24 bolts. Install 3/8-in. diameter dowels. Tighten 3/8-24 bolts to 45 ft-lb.
11. Install shaft seals into appropriate housing.
12. Prime all end and seal cap mating surfaces (both component parts and transmission housing) with "Loctite" Primer H.
13. Apply a 1/16-in. diameter bead of "Loctite" gasket eliminator No. 5155 to the flanged surfaces of all the end and seal caps as shown below.



14. Install the covers and seal housing. Torque bolts to 50 in.-lb.

15. Install upper shaft coupling hubs as per Section 4.6.3, Steps 4 and 5.

16. Install transmission into test stand by reversing the removal procedure of Section 4.1.

17. Torque hold-down bolts to 50 ft-lb.

5.0 SPINDLES

5.1 Removal

1. If discs are installed, remove them (see Part I, Operating Instructions, Section 2.3).
2. Remove cooler clamp (Item 17, Drawing 759J003).
3. Remove test head bolts and taper dowels (Items 35 and 23, respectively, Drawing 759E066).
4. Disconnect test oil supply line.
5. Remove heater power wires. If installed, make sure heating elements will clear test lube oil system when the test head is removed.
6. Disconnect thermocouples.
7. Carefully slide test head housing off guide pins.
8. Place test head on clean surface, gasket side down.
9. Remove test head bracket bolts and taper pins (Items 65 and 33, respectively, Drawing 759J003) and remove test head bracket assembly (Item 2, Drawing 759E066).
10. Remove coupling bolts from coupling hub on spindle.
11. Remove spindle hold-down bolts and taper pins (Items 62 and 34, respectively, Drawing 759J003).
12. Remove spindle(s). Remove load cell if required. Do not disturb shims.

5.2 Replacement

1. Wipe mounting surfaces with clean cloth and position spindle over mounting holes.
2. Install spindle hold-down bolts and taper pins (Items 62 and 34, respectively, Drawing 759J003). Torque bolts to 30 ft-lb minimum.
3. Install test head bracket, bolts, and taper pins. Torque bolts to 50 ft-lb minimum.
4. Wipe test head mounting surfaces with clean lint-free cloth.
5. Install seal positioning cups (SK-B-8805) onto spindle shafts.
6. Carefully position test head on alignment dowels and push into position.
7. Install and hand-tighten the four test head bolts.
8. Install two tapered dowels.
9. Torque test head bolts to 50 ft-lb.
10. Reconnect oil supply line.
11. Reconnect thermocouples.
12. Install cooler clamp (Item 17, Drawing 759J003) with new gasket (Item 35, Drawing 759J003).
13. Reinstall coupling bolts.

5.3 Disassembly

> **NOTE**
>
> *The internal structures of the movable spindle and the fixed, electrically insulated spindle are different.*

5.3.1 Movable Spindle

Refer to Drawing 759D064G1.

1. Remove spindle shaft coupling hubs with a suitable puller (see Section 4.3.1, Step 2). Protect the shaft center during this operation. Maintain hub/shaft identification.
2. Remove end cap (Item 5).
3. Remove bearing lock nut (Item 18) and lock washer (Item 17).
4. Remove cap screws (Item 19) from front end cap.
5. Support spindle housing (Item 1) and press shaft out of housing, taking the larger ball bearings with it. Protect the shaft center during this operation.
6. With a suitable puller, remove the remaining shaft-mounted ball bearings (Item 14) and spacer (Item 6).
7. Remove all other spindle components.
8. Discard all bearings.

5.3.2 Fixed, Electrically Insulated Spindle

Refer to Drawing 759D064G2.

1. Remove end cap (Item 5).
2. Remove cap screws (Item 19) from front end cap.
3. Gently push shaft cartridge assembly out of the housing.
4. Remove bearing locknut (Item 18) and locknut washer (Item 17).
5. With a suitable puller pressing against the outermost ball bearing (Item 14), push the shaft out of all bearings and sleeves.
6. Discard all bearings.

5.4 Assembly

> **NOTE**
>
> *The internal structures of the movable spindle and the fixed, electrically insulated spindle are different.*

5.4.1 Movable Spindle

Refer to Drawing 759D064G1.

1. Remove one shield from each bearing to be installed (two of Item 14, two of Item 15, Drawing 759D064).
2. Clean bearings using normal cleaning procedures.
3. Repack each Size 210 bearing with 6.5 to 7 gm of Mobil SHC 32 grease and each Size 109 bearing with 5.3 to 5.5 gm of Mobil SHC 32 grease.
4. Clean remaining spindle components.
5. Place two bearings (Item 14), shield side down, and spacer (Item 6) in oven and heat at 250°F for 1 hr.
6. Install one Size 210 bearing (Item 14) with shield side down onto shaft. Be sure it is seated.
7. Install second Size 210 bearing (Item 14) with shield side up onto shaft. Be sure it is seated.
8. Install spacer (Item 6) onto shaft. Be sure it is seated.
9. Place washer (Item 7) into housing.
10. Place compression springs (Item 16) into spring guide (Item 8). The springs can be held in position with a slight amount of grease; then install spring guide into housing.
11. With shaft in vertical position, lower housing over shaft until it is seated against bearings. Do not use excessive force.
12. Secure end cover (Item 4) to housing with cap screws (Item 19).
13. With shaft vertical, install one Size 109 bearing (Item 15), shield side down, onto shaft. The bearing is tight on the shaft and loose in the housing and will require some force to install; push only on inner race.
14. Install second Size 109 bearing (Item 15), shield side up, onto shaft.
15. Install lockwasher (Item 17) and locknut (Item 18). Tighten locknut hand tight.
16. Install end cover (Item 5) and cap screws (Item 20).
17. Restrain shaft and tighten locknut. Be careful not to damage test disc mounting surface and sealing surface on heat dam. Secure locknut by bending appropriate lockwasher tab.

5.4.2 Fixed Spindle

Refer to Drawing 759D064G2.

1. Remove one shield from each bearing to be installed (two of Item 14, two of Item 15, Drawing 759D064).
2. Clean bearings using normal cleaning procedures.
3. Repack each bearing with 6.5 to 7 gm for the 210 size, 5 1/3 to 5 1/2 gm for the 209 size Mobil SHC 32 grease.
4. Clean remaining spindle components.

5. Place two bearings (Item 14), shield side down, and spacer (Item 6) in oven and heat at 250°F for 1 hr.
6. Install one Size 210 bearing (Item 14) with shield side down onto shaft. Be sure it is seated.
7. Install second Size 210 bearing (Item 14) with shield side up onto shaft. Be sure it is seated.
8. Install spacer (Item 6) onto shaft. Be sure it is seated.
9. With the shaft vertical, place housing spacer (Item 10) onto shaft.
10. Place washer (Item 7) onto spacer.
11. Put compression springs (Item 16) into spring guide (Item 8) and place over washer. The springs can be held in position with a small amount of grease.
12. Heat two Size 109 bearings (Item 15) for 1 hr.
13. Install one heated Size 109 bearing (Item 15) onto shaft, shield side down. Provide axial force against inner race to seat bearing against shoulder. (This process will compress the springs (Item 16).)
14. Install second heated Size 109 bearing (Item 15) onto shaft. Provide axial force against inner race to seat bearing until it is cool.
15. Install lockwasher (Item 17) and locknut (Item 18). Tighten locknut hand tight.
16. Make sure the shaft-mounted parts are aligned concentrically.
17. Lower the shaft assembly into the housing until the bearings (Item 14) are seated.
18. Install end cover (Item 5) and cap screws (Item 20).
19. Restrain shaft and tighten locknut. Be careful not to damage test disc mounting surface and sealing surface on heat dam. Secure locknut by bending appropriate lock washer tab.